JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXVII. NOVEMBER, 1915. No. 4.

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THE MYSTERY OF THE SCREW PROPELLER.

BY CAPTAIN C. W. DYSON, U. S. NAVY, MEMBER.

For many years after the adoption of the screw propeller for the propulsion of ships the seeming vagaries in its performances in actual service of propulsion have cast a mystery over it and over the laws governing its action.

The greater part of this mystery is, however, not due to the propeller, but can be directly placed on the carelessness with which trials of ships have been run and data collected, while the greater part of the remainder can be attributed to the effect of variations in hull form in interfering with the proper flow of water to the propeller, thus seriously decreasing the propulsive efficiency; in producing more or less wake and so adding to the propulsive efficiency; in incorrect estimates of effective horsepowers required for given speeds, basing these powers on estimates of frictional and residual resistances with an entire neglect of the malign influence of the appendages such as struts, bilge keels, etc.

The small remaining portion of the mystery can be ascribed

to the propeller itself, and is due to the myriad variations in blade forms and sections which have been used, these appearing to depend upon the taste of the designer, and upon the lack of a consistent basis of comparison for the analysis of propeller performances.

As the years rolled by they brought in their wake the model tank by which a more nearly accurate value of the effective horsepower can be obtained; more accurate instruments for the measurement of indicated and shaft horsepowers; better mechanical construction of propelling engines by which frictional losses have been greatly reduced and brought to a more nearly constant and even value; more care in running trials over the measured mile combined with a better knowledge of the effect of shallow water on the course, both tending towards the production of better speed data; and these improvements have resulted in the production of data from which curves of analysis may be drawn, rendering it possible to take any propeller which may be proposed for any given vessel, and to estimate a curve of performance for any condition of loading with more than a fair degree of accuracy.

K—CORRECTIVE FACTOR FOR INDICATED AND SHAFT HORSEPOWERS.

When a propeller works at the stern of a vessel it operates in a body of water which partakes, to a more or less degree, of the forward movement of the vessel itself. When the propeller is so located that the suction column of water entering the propeller disc enters normal to the disc and with very little disturbance, and when the propeller blade tips are well immersed and pass the hull at a good distance from it, the wake will increase the effective thrust of the propeller, and therefore the effective horsepower delivered, for any given indicated or shaft horsepower which may be delivered by the propelling engine. This gain is known as the *wake gain.*

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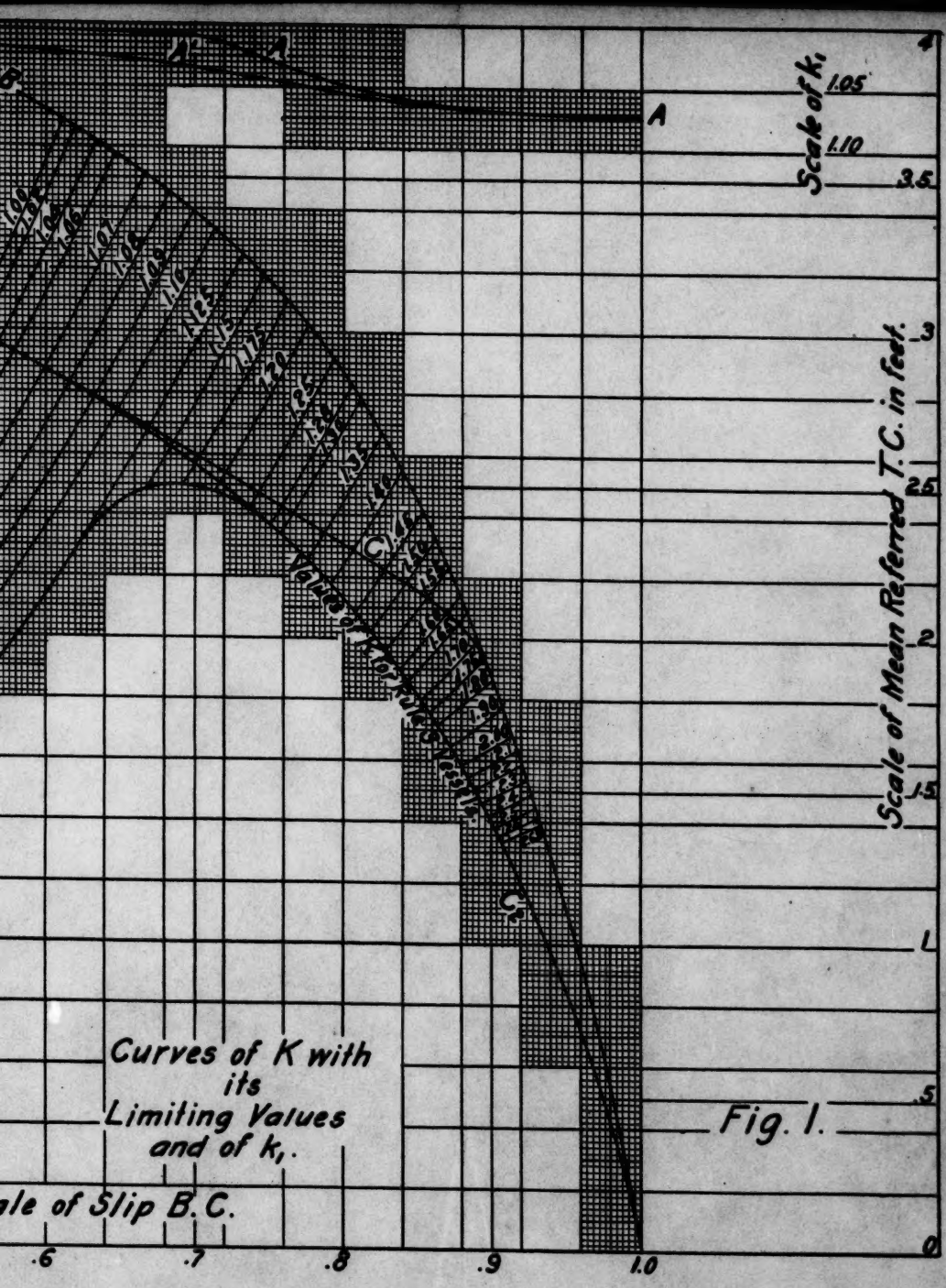
k_1 { A-A - Center Screw Referred T.C. less than 3ft.
A'-A- " " " T.C. greater than 3ft.
A'-A - Propellers located on wing shafts.

K { B - Approximate minimum value
C-C_2 - Approximate maximum values for center screws and for wing propellers of vessels under Rule 3.
C'-C_2 - Wing propellers for vessels under Rule. 1.
C-C - Minimum values of K for vessels under Rule. 3.

In no case should K be taken less than k_1.



Curves of K with its Limiting Values and of k_1.

Fig. 1.

ale of Slip B.C.

that the suction column is no more a cylinder but becomes a frustrum of a cone, the propeller disc being the upper bounding plane, or should the propeller blades be insufficiently immersed or pass unduly close to the hull, or should combinations of the above conditions exist, the effective thrust, and consequently the effective horsepower, per revolution for any indicated or shaft horsepower delivered by the propelling engine, will be reduced. This loss is commonly called the *thrust deduction.*

In cases where the thrust deduction exceeds the wake gain, and such cases are the usual ones, the result is a net loss in propulsive efficiency and an increase in revolutions for the given speed.

Where the wake gain exceeds the thrust deduction, and such cases are rare, the opposite effects are produced.

The factor K represents unity plus the net loss or minus the net gain, as the case may be.

CONTROL OF THE VALUE OF K.

As the value of K is fixed by the lines of the hull and the relative positions of propeller and hull, there exists a certain amount of freedom in fixing the loss due to this relative position. By practical considerations which are forced, the propeller cannot be further removed aft from the fullness of the lines than a certain amount for any hull, these considerations being governed by the necessity for the shaft and propeller supports, and this maximum distance, fore and aft, fixes the minimum value of K for any hull. This value of K will then be increased as the tip clearance between hull and propeller decreases below a certain amount, this amount depending upon the slip block coefficient and upon the height of the horizontal line of least tip clearance, usually the height of the hub center above the base line of the vessel. The amount of immersion of the upper tips of the blades below the water surface appears to have some influence, but very slight as compared with

tip clearance for vessels acting on the surface. In the case of submarines, however, where the hydraulic head becomes very much increased by diving, the depth of immersion increases in importance and undoubtedly adds to the *propulsive efficiency* of the propeller.

The full value of K is made up of two quantities, which may be called k_1 and k_2. k_1 is the natural loss due to the hull itself, affecting the density of the water in the column flowing to the propeller, the propeller being well located and with good tip clearance. k_1, on Fig. 1, is represented by the curve marked A. k_1 requires increased power and increased revolutions to deliver any given effective horsepower. k_2, on the other hand, affects the indicated or shaft horsepower only, and is caused by the malign influence of currents of water entering the propeller disc radially and to eddying between the blades, neither adding anything to the thrust but adding directly to the resistance to turning. Therefore, in estimating revolutions for any given speed, k_1, only, is used, as also in computing diameter of the propeller, while for the estimate of final I.H.P. or S.H.P., the full value of $K = k_1 + k_2$ must be used.

On this Figure are shown curves of K, having slip block coefficients as abscissas and mean blade-tip clearances as ordinates. On this same figure are shown cross curves giving the minimum, marked B and maximum C^1 values of K that can be expected with blades of Standard Form and with Standard Hull Form. There are also shown curves of k_1 for propellers located under the quarters of a vessel as well as for those located directly in wake of the stern post.

The term "mean tip clearance" is misleading, as the ordinates are averages of tip clearance and immersion of upper blade tip, both being referred to standard conditions of clearance and immersion and the mean taken by giving each factor its apparent weight in the equation for "mean tip clearance."

This equation is:

Mean tip clearance $= \frac{H + 10\ H^1}{11}$, where

H^1 (Twin, 4, etc., screws) = (Actual horizontal T.C. $\times$ 10) $\div$ Actual height of center of hub above base line.

H^1 (Single or Center Screw) = Fore-and-aft Clearance in feet between center line of vertical blade and fullness of hull.

H (depth of immersion of upper blade tip) = (Actual Immersion in feet $\times$ 14) $\div$ actual height of tip above base line.

RULES FOR APPLYING THE ABOVE.

RULE 1.—*Where blades are of standard form, and in the cases of other than center-line propellers when a vertical line through the propeller hub center penetrates the hull close to or below the load-water plane, apply the curves as they stand.*

RULE 2.—*Where the blades are narrow tipped, as mean tip clearance passes from max. K to min. K, decrease values of K by from .03 at max. to 0 at min. Where broad-tipped or bulbous, increase K by same amounts.*

RULE 3.—*When vertical line through hub center, with other than center-line propellers, pierces hull well above the load-water plane, use curve of K marked C and do not correct slip block coefficient for variation of midship section coefficient.*

THE SCREW PROPELLER.

All screw propellers when working under similar conditions of resistance arrange themselves in one great family in which the position of any particular propeller is fixed by its diameter, its pitch and its projected area ratio, the latter fixing the dimension of the thrusts and the resultant tip speed and propulsive efficiency which may be expected.

As the condition of the resistance changes the values of the thrusts and of the tip-speed change in inverse proportion.

These thrusts, tip speeds and propulsive efficiencies can be shown graphically as curves, the abscissa values of which are

projected area ratios, and the family position of any propeller can be found by using these values to ascertain the indicated or shaft horsepower and the effective horsepower which the propeller will deliver under the conditions of resistance which produce such curves. These conditions will be spoken of as "Chart Conditions."

On Fig. 2 are shown the curves of propulsive coefficients, tip speeds in feet per minute, and indicated thrusts in pounds per square inch of disc area of the propeller for values of projected area ratios from zero to over .6, and may be regarded as accurate from the value .25 to the maximum.

These curves are exact reproductions of the corresponding design curves shown on Chart 5 (Sheet 21) of the author's work on "Screw Propellers" except the portion of the propulsive coefficient curve from a projected area ratio of .56 to the maximum ratios shown; experience in this range has shown conclusively that higher values of propulsive efficiency can be realized in this region than were credited to it in the original curve, and the curve has been corrected and brought up to the values which may be confidently looked for unless peculiarities of hull, abnormal trial conditions, or position of propeller intervene and produce conditions of which no account can be taken in making a forecast of a propeller's performance.

DATA AND FORMULAS FOR FINDING THE CHART CONDITION OF A PROPELLER.

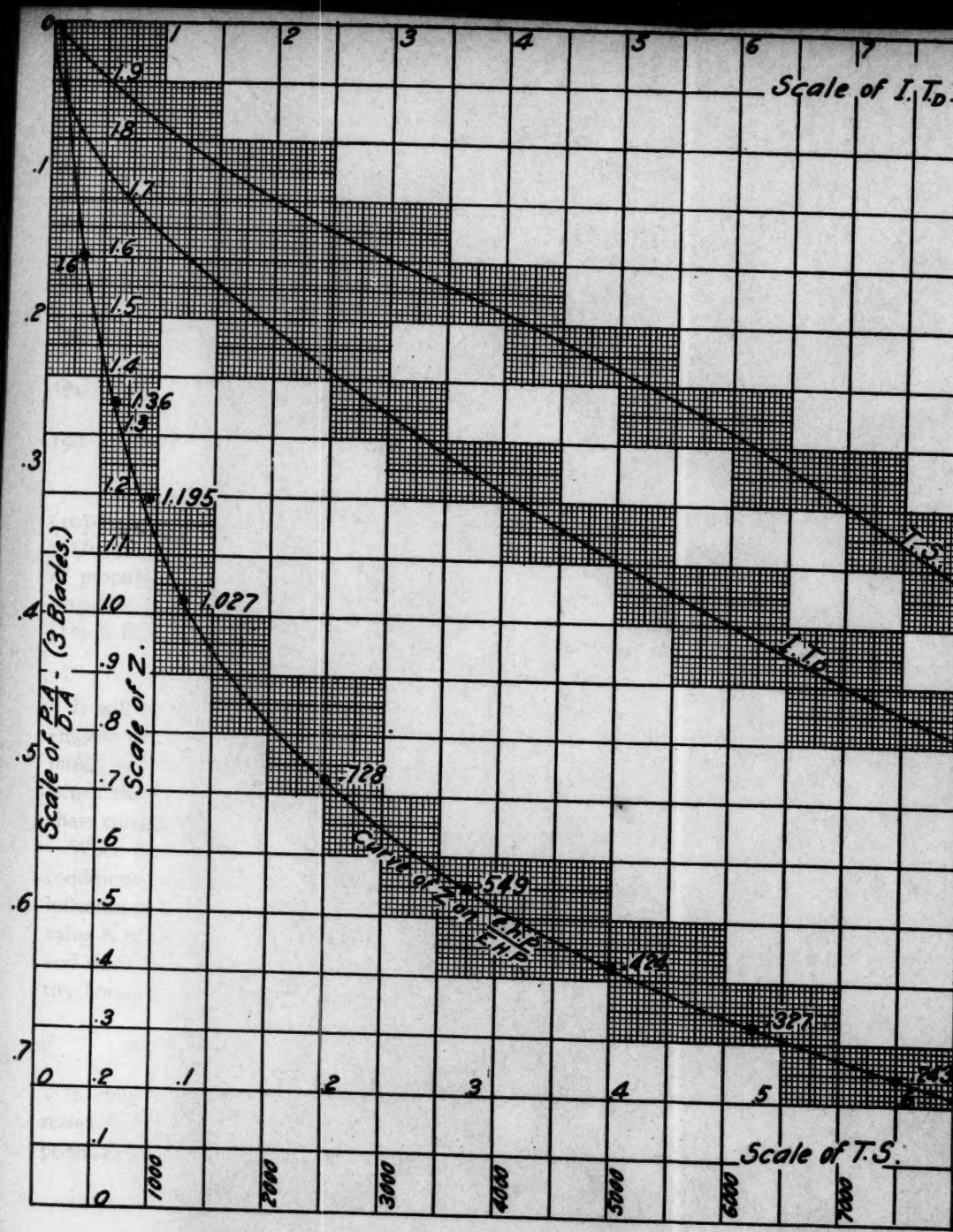
In finding the chart condition of any given propeller the following data must be at hand:

(1) P.A. ÷ D.A. = Actual projected area ratio of a three-bladed, $\frac{4}{3}$ of this ratio for a 4 and $\frac{3}{2}$ of it for a 2-bladed propeller.

(2) P = Pitch of the propeller, in feet.

(3) D = Diameter of the propeller, in feet.

(4) T.S. = Tip speed for P.A. ÷ D.A. (Fig. 2).



Fig

NOTE.—On curve of z, every tenth value of $\frac{e.h.p.}{E.H.P.}$ is indicated by a circ

Fig. 2.

ted by a circle and the empirical values of z at these points are given.

(5) R = T.S. ÷ πD = Revolutions under chart conditions.
(6) P × R = Pitch in feet × revolutions per minute.
(7) I.T.$_d$ = Indicated thrust per square inch of disc area or P.A ÷ D.A. (Fig. 2).
(8) N = Number of propellers working under similar conditions.
(9$_a$) I.H.P. = (N × D^2 × I.T.$_d$ × P × R) ÷ 291.8 = indicated horsepower (3 blades).
(9$_b$) I.H.P. = (N × D^2 × I.T.$_d$ × P × R) ÷ (291.8 × .865) = indicated horsepower (4 blades).
(9$_c$) I.H.P. = (.75 × N × D^2 × I.T.$_d$ × P × R) ÷ 291.8 = indicated horsepower (2 blades) all under chart conditions.
(10) P.C. = Propulsive coefficient for P.A. ÷ D.A. (Fig. 2).
(In the cases of four and two-bladed propellers, the value of propulsive coefficient given by Fig. 2 for the actual full projected area ratio must be used.)
(11) E.H.P. = I.H.P. × P.C. = effective horsepower delivered by the propeller under chart conditions.

It will be noted at once that the value of the slip block coefficient of the vessel has no place whatsoever in these formulas, nor does it have any effect of any kind on the work which the propeller can perform when operating under the chart condition.

When the chart condition is departed from and the actual conditions existing in the wake of the hull are considered, the influence of the hull form must be taken into account and the value K is finally brought into the equation for final indicated and shaft horsepower necessary to deliver any value of effective horsepower.

ESTIMATE OF CHART CONDITION OF SPEED, V.

In obtaining this estimate, the slip block coefficient (corrected for midship section coefficient and for position of propeller, as explained in the author's work, "Screw Propellers"),

enters into the data values, as this coefficient affects the value of the effective thrust per square inch of projected area, and the factor 1 — S, in which S is the apparent slip for the particular slip block coefficient and projected area ratio under consideration, is equal to the propulsive thrust in pounds per square inch of projected area divided by the effective thrust. The data and formulas for obtaining V are:

(12) $P.T._p$ = Propulsive thrust in pounds per square inch projected area for P.A. ÷ D.A. (Chart 5, sheet 21, "Screw Propellers").

(13) $E.T._p$ = Effective thrust in pounds per square inch projected area for P.A. ÷ D.A. and for slip block coefficient (Chart 5, sheet 21, "Screw Propellers").

(14) S = Apparent slip under chart conditions.

(15) $1 - S = P.T._p \div E.T._p$ = per cent. advance of ship under chart conditions.

(16) $V = \left\{ P \times R \times (1 - S) \right\} \div 101.33$ = speed of ship under chart conditions.

TO CHANGE FROM CHART CONDITIONS TO OTHER CONDITIONS OF RESISTANCE.

Suppose a vessel to be so loaded that for the speed V an indicated horsepower I.H.P., or a shaft horsepower, S.H.P., are required to deliver the effective horsepower, E.H.P., necessary to produce this speed, the revolutions of the propeller being R and the tip speed T.S. The propeller is then working under chart conditions of resistance.

Should the speed be reduced by reducing the power of the engines, by increasing the displacement, by fouling of bottom, by condition of wind or sea, etc., or should the opposite effect occur and the speed be increased, the conditions of resistance differ from the chart conditions and the following changes occur in the propeller performance:

Chart Condition of Loading for V, but

Engine power reduced:—
Effective horsepower reduced.
Revolutions reduced.
Tip speed reduced.
Thrusts reduced.
Speed reduced from V to *v*, I.H.P. reduced to I.H.P. and where K is greater than unity, to K × I.H.P.$_a$.

Load increased for V above Chart Condition.

Chart engine power constant = I.H.P.:—
Effective horsepower = E.H.P.
Revolutions reduced.
Tip speed reduced.
Thrusts increased.
Speed reduced from V to *v*, and where K is greater than unity I.H.P. increases to K × I.H.P.

Load decreased for V below Chart Conditions.

Chart engine power constant = I.H.P.:—
Effective horsepower = E.H.P.
Revolutions increased.
Tip speed increased.
Thrusts decreased.
Speed increased from V to *v*, and where K is greater than unity I.H.P. becomes K × I.H.P.

It will be noted in these changes of conditions that so long as the I.H.P. remains constant the E.H.P. remains constant also, no matter how the actual speed of the ship may change. This statement is borne out by comparison of trial results of several vessels where the vessels were of sufficiently fine after body and propellers were so well placed as to practically assure that K equalled unity. In these cases the agreement between

the actual indicated, shaft, and effective horsepowers, and those of the chart condition of the propeller were so close as to justify the following statement:

RULE 4.—*Should a screw propeller working in the wake of a vessel deliver a certain effective horsepower with a certain indicated or shaft horsepower under any given condition of resistance, it will deliver the same effective with the same indicated or shaft horsepower under any other condition of resistance so long as it is operating in the wake of the same hull, and so long as the effective thrusts are below the point of cavitation.*

This law renders it possible, where a vessel has been tried up to and beyond the speed for which the effective horsepower is equal to the Chart Condition E.H.P. of the propeller used, to obtain the value of K at once as—

The actual indicated horsepower required to deliver E.H.P. = K × I.H.P. (Chart Condition), from which K at once results

$$K = \frac{\text{actual indicated horsepower}}{\text{I.H.P. (Chart Condition)}} = \frac{\text{Actual S.H.P.}}{.92\ \text{I.H.P. (Chart cond.)}}$$

DETERMINATION OF POWER FACTOR Z.

In arriving at a satisfactory series of values of condition factors to use in estimating power due to changes in conditions of resistance from the Chart Conditions, many different forms of equations were tried, using the measured-mile trial data of very long vessels, tried in deep water, and where the trials were conducted in such a manner as to create confidence in the trial results tabulated. Thus all the trials that have been used have had at least three runs for each plotted point of the speed-revolution and speed-power curves while the highest point of the curve has been obtained by five runs. In obtaining the mean of each set of runs, the following method of averaging was used:—

For a three-run point:

Run No. 1.	North.	1 × Power.	1 × Revolutions.
Run No. 2.	South.	2 × Power.	2 × Revolutions.
Run No. 3.	North.	1 × Power.	1 × Revolutions.
Mean, ...		$\frac{\text{Sum}}{4}$	$\frac{\text{Sum}}{4}$

For a five-run point:

Run No. 1.	North.	1 × Power.	1 × Revolutions.
Run No. 2.	South.	2 × Power.	2 × Revolutions.
Run No. 3.	North.	2 × Power.	2 × Revolutions.
Run No. 4.	South.	2 × Power.	2 × Revolutions.
Run No. 5.	North.	1 × Power.	1 × Revolutions.
Mean, ...		$\frac{\text{Sum}}{8}$	$\frac{\text{Sum}}{8}$

The final form of the estimating power equation became:

$$\text{I.H.P.}_d = \text{I.H.P.} \left(\frac{v}{V}\right)^w,$$

where v is any speed and I.H.P.$_d$ the indicated horsepower for this speed. When K for the vessel under consideration exceeds unity, the actual indicated horsepower for $v = K \times$ I.H.P.$_d$.

Designating the effective horsepower necessary to obtain the speed v, by *e.h.p.*, and taking values of v for several trial vessels for the load ratios $\frac{e.h.p.}{E.H.P.} = .025, .05, .075, .1, .2, .3, .4$, etc., up to 1.1, and solving the equation

$$w = (\log \text{I.H.P.} - \log \text{I.H.P.}_d) \div (\log V - \log v),$$

I.H.P.$_d$ being the actual indicated horsepower for v, where $K = 1$, and being equal to the actual indicated horsepower for v divided by K where $K > 1$, a series of curves are obtained for the different load ratios given.

Taking $z = w(\log V - \log v)$, it was found that for each curve of w, depending on the load ratio $\frac{e.h.p.}{E.H.P.}$, z had practically a constant value.

These values of z are shown as a curve on Fig. 2, having the ratios $\frac{e.h.p.}{E.H.P.}$ as abscissas, and they vary in value from ∞ when $\frac{e.h.p.}{E.H.P.} = 0$, to zero when $\frac{e.h.p.}{E.H.P.} = 1$, and then again increase in value as $\frac{e.h.p.}{E.H.P.}$ becomes greater than unity. In this latter case, however, z changes sign. The power equation therefore can be simplified to the forms:

(*a*) . . $\frac{e.h.p.}{E.H.P.} < 1$, $\log I.H.P._d = \log I.H.P. - z$, z being positive.

(*b*) . . $\frac{e.h.p.}{E.H.P.} > 1$, $\log I.H.P._d = \log I.H.P. - z$, z being negative.

and the final indicated horsepower for any speed v becomes $= K \times I.H.P._d$, and is absolutely independent of speed but depends entirely upon the load ratio $\frac{e.h.p.}{E.H.P.}$ for its value.

While w is of no further use in obtaining the estimate of power, z having usurped its place, it is of interest to obtain its values for different values of $\frac{v}{V}$ for varying ratios of $\frac{e.h.p.}{E.H.P.}$, and then to plot the values corresponding to each value of $\frac{e.h.p.}{E.H.P.}$ as a curve, using the corresponding values of $\frac{v}{V}$ as abscissas.

There results from such plotting, two complete series of rectangular hyperbolas, Fig. 3, one for conditions where

$$\frac{e.h.p.}{E.H.P.} < 1 \text{ and one where } \frac{e.h.p.}{E.H.P.} > 1,$$

the rectangular asymptotes intersecting each other at

$$\frac{v}{V} = 1.$$

Thus, in the first quadrant and the third quadrant, opposite branches of the same hyperbolas, there will be

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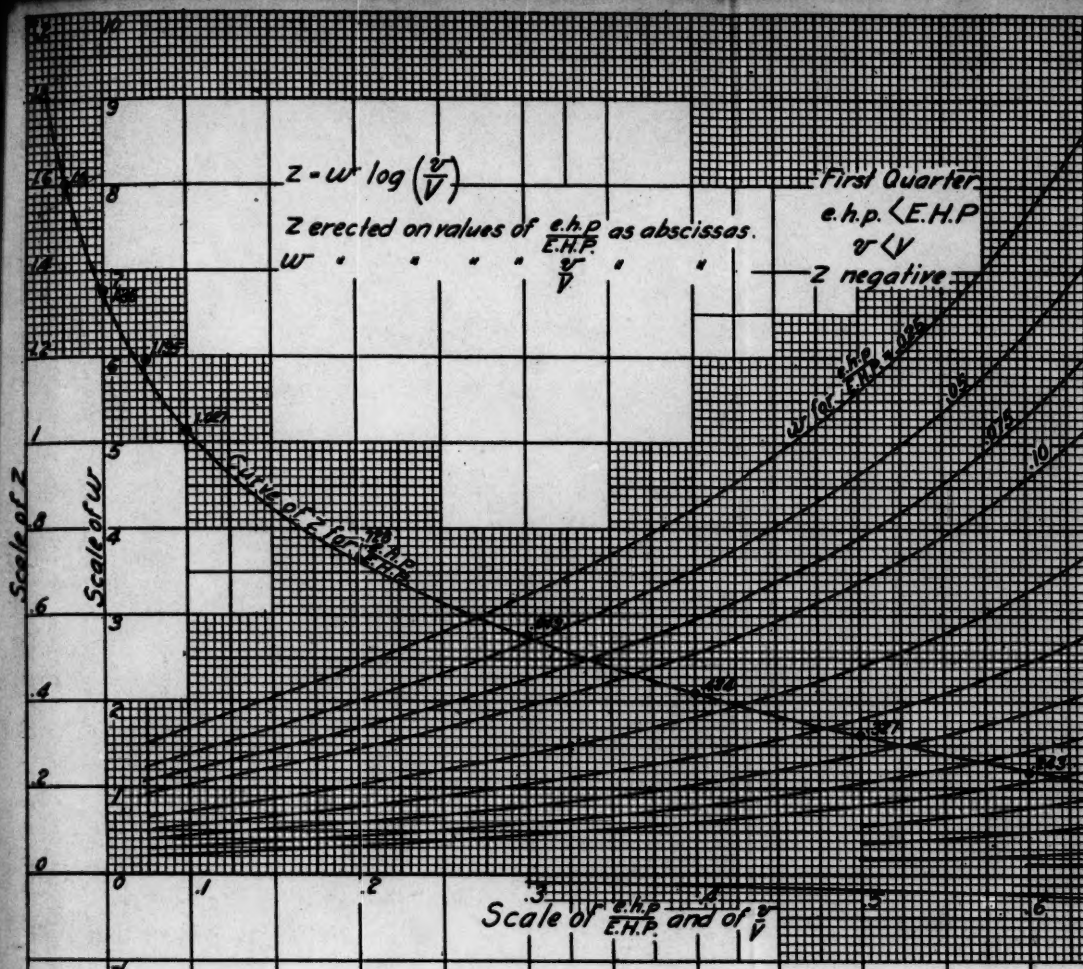
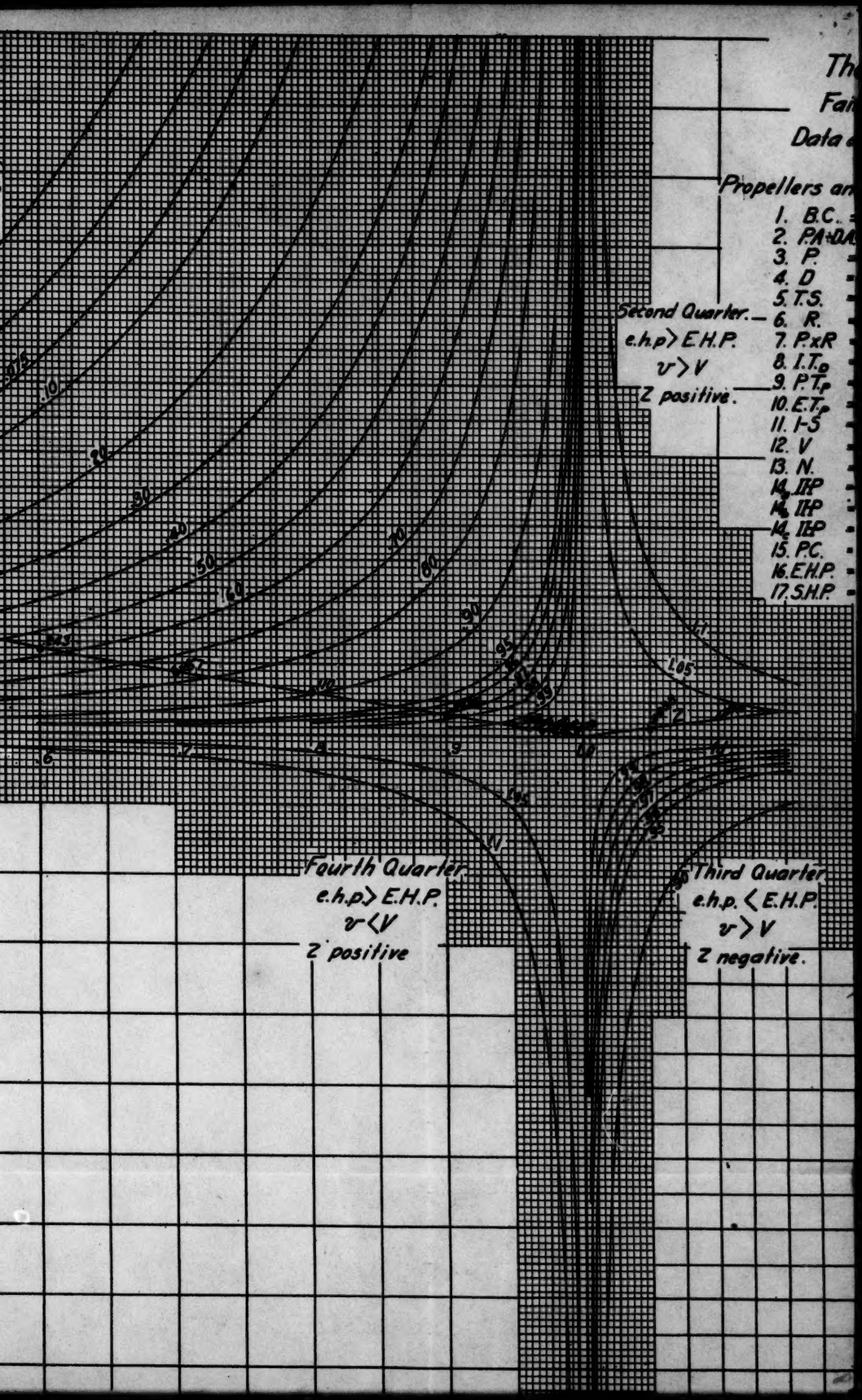

$z = w \log\left(\frac{v}{V}\right)$
z erected on values of $\frac{e.h.p}{E.H.P.}$ as abscissas.
w " " " " $\frac{v}{V}$ " "
First Quarter.
e.h.p. < E.H.P
v < V
z negative.
Scale of z
Scale of w
Scale of $\frac{e.h.p}{E.H.P.}$ and of $\frac{v}{V}$
Scale of w

Forms for Analysis.

18. v = desired speed
19. e.h.p. = Effective Horse Power for v.
20. $\frac{e.h.p.}{E.H.P}$ = load factor.
21. z = value for $\frac{e.h.p}{E.H.P.}$
22. $\log IHP_d = \log IHP \pm z$ = Primary Power for v
23. K = Corrector for B.C. and Tip Clearance.
24. Final $IHP_d = K \times IHP_d$.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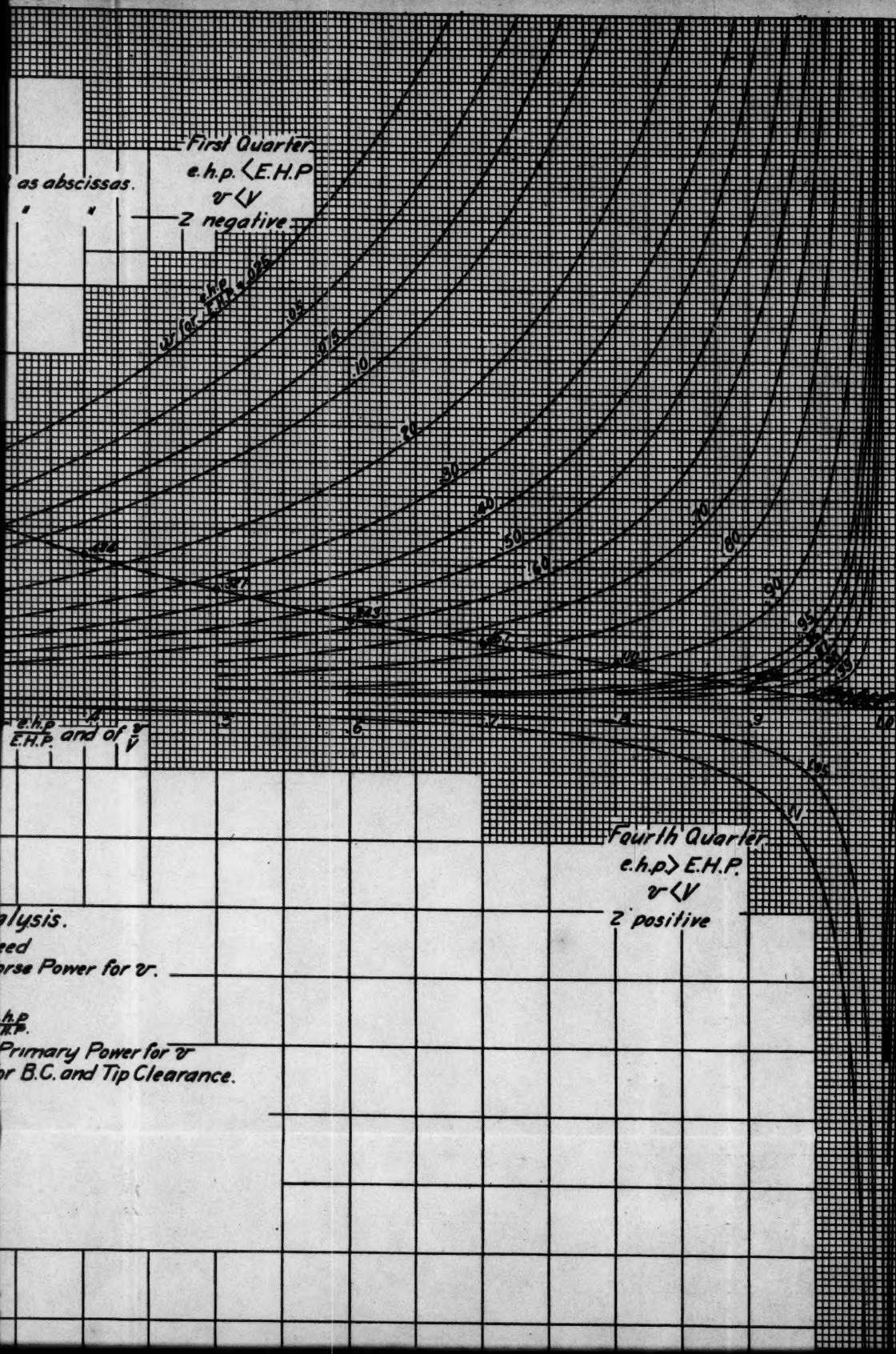
Data
Propellers an
1. B.C.
2. P.A.-D.A.
3. P.
4. D
5. T.S.
6. R.
7. P.xR
8. I.T.
9. P.T.
10. E.T.
11. I-S
12. V
13. N.
14. IHP
14. IHP
14. IHP
15. P.C.
16. E.H.P.
17. S.H.P.
Second Quarter.
e.h.p > E.H.P.
v > V
Z positive.
Fourth Quarter.
e.h.p > E.H.P.
v < V
Z positive
Third Quarter
e.h.p. < E.H.P.
v > V
Z negative.


First Quarter.
e.h.p. < E.H.P
v < V
z negative
as abscissas.
Fourth Quarter.
e.h.p > E.H.P.
v < V
z positive
Primary Power for v
B.C. and Tip Clearance.

The Mystery of the Propeller.

Faired Curves of w and z.

Data and Formulas for Analysis of Propellers and Estimate of their Performance.

1. B.C. = Slip Block Coefficient of vessel (Chart 1)
2. P.A÷D.A. = Actual Proj. Area Ratio of 3 blades.
3. P. = Pitch of Propeller in feet.
4. D = Diam. " " "
5. T.S. = Tip Speed of Propeller for P.A.÷D.A. (Chart 5)
6. R. = T.S. ÷ πD = Revs. under Chart Conditions.
7. P.xR =
8. $I.T_o$ = Ind. Thrust for P.A.÷D.A. (Chart 5)
9. $P.T_p$ = Propulsive " " " " " (" 5)
10. $E.T_p$ = Effective " " " " " (" 5)
11. I-S = $P.T_p - E.T_p$ = I- Apparent Slip (Chart Cond.)
12. V = PxR x (I-S) ÷ 101.33 = Speed, " "
13. N. = Number of Propellers operating under similar conditions.
14. $I.H.P._a$ = (N x D^2 x $I.T_o$ x PxR) ÷ 291.8 = Ind. Horse Power 3 blades.
14. $I.H.P._b$ = (N x D^2 x $I.T_o$ x PxR) ÷ (291.8 x .865) " " 4 "
14. $I.H.P._c$ = (N x D^2 x $I.T_o$ x PxR x .75) ÷ 291.8 = " " 2 "
15. P.C. = Propulsive Coefficient for P.A.÷D.A. (Chart 5)
16. E.H.P. = P.C x I.H.P = Effective Horse Power (Chart Condition.)
17. S.H.P. = I.H.P x .92 = Shaft Horse Power.

Second Quarter.
e.h.p > E.H.P.
$v > V$
Z positive.

Third Quarter
e.h.p. < E.H.P.
$v > V$
Z negative.

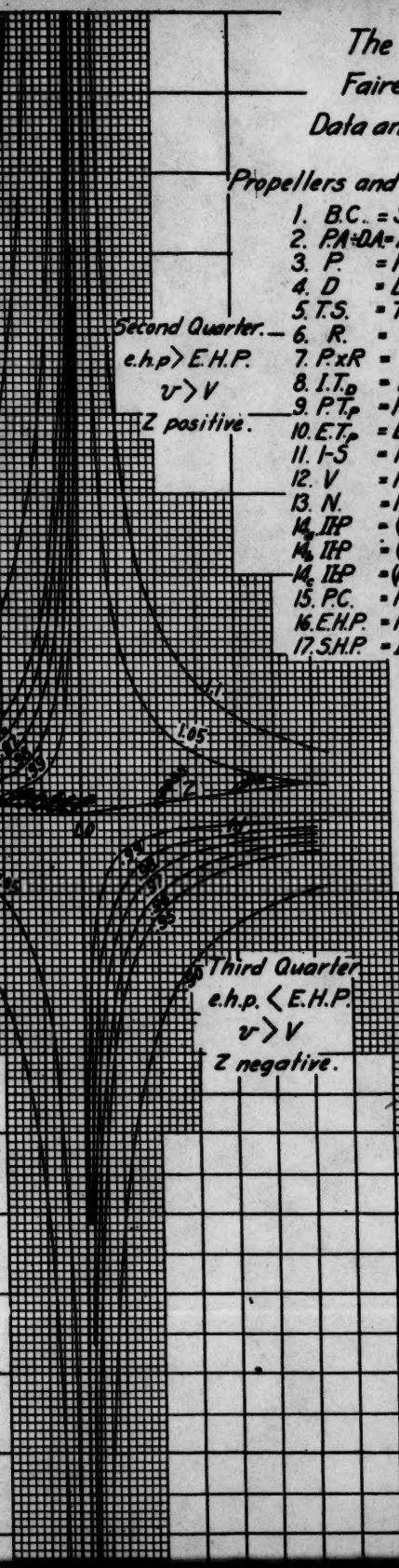
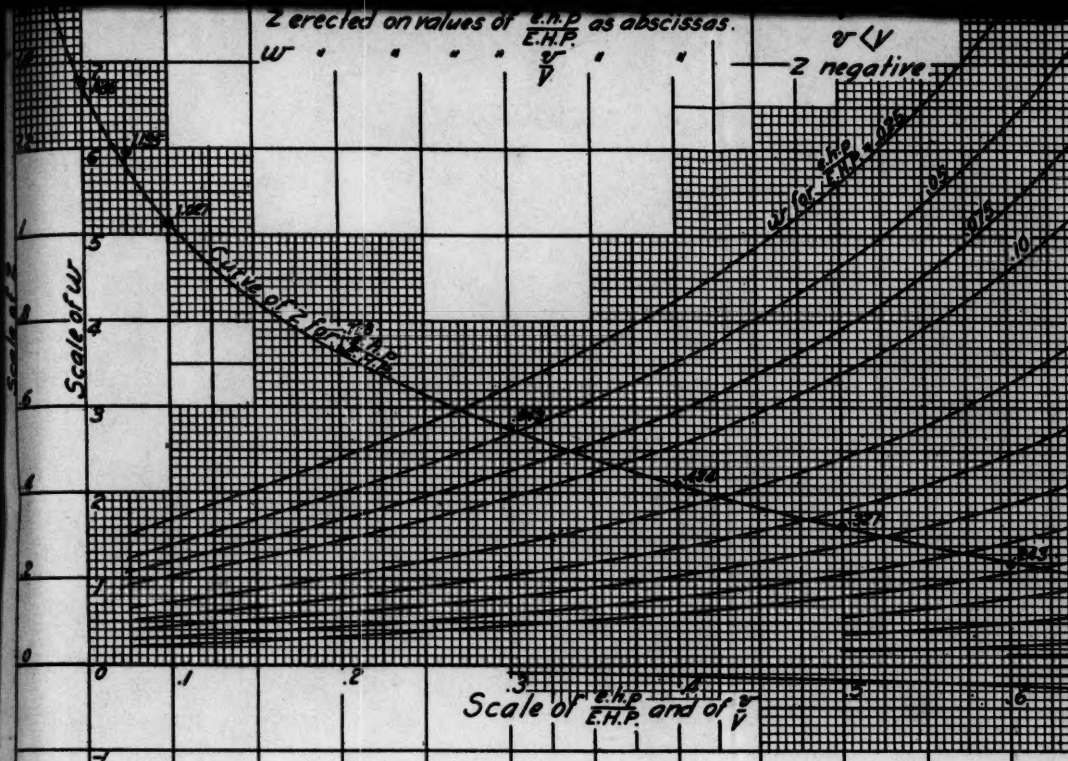


Fig. 3



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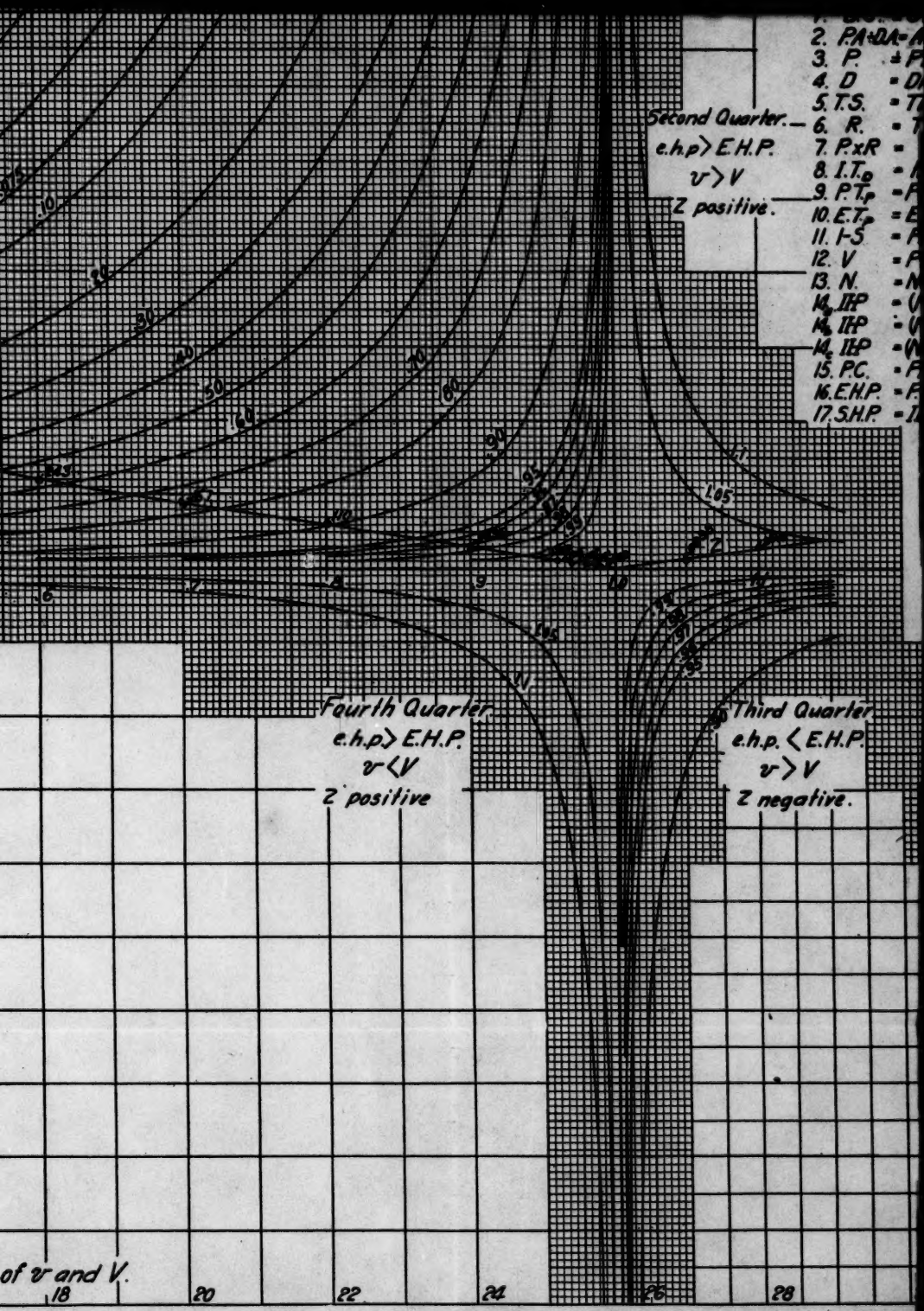
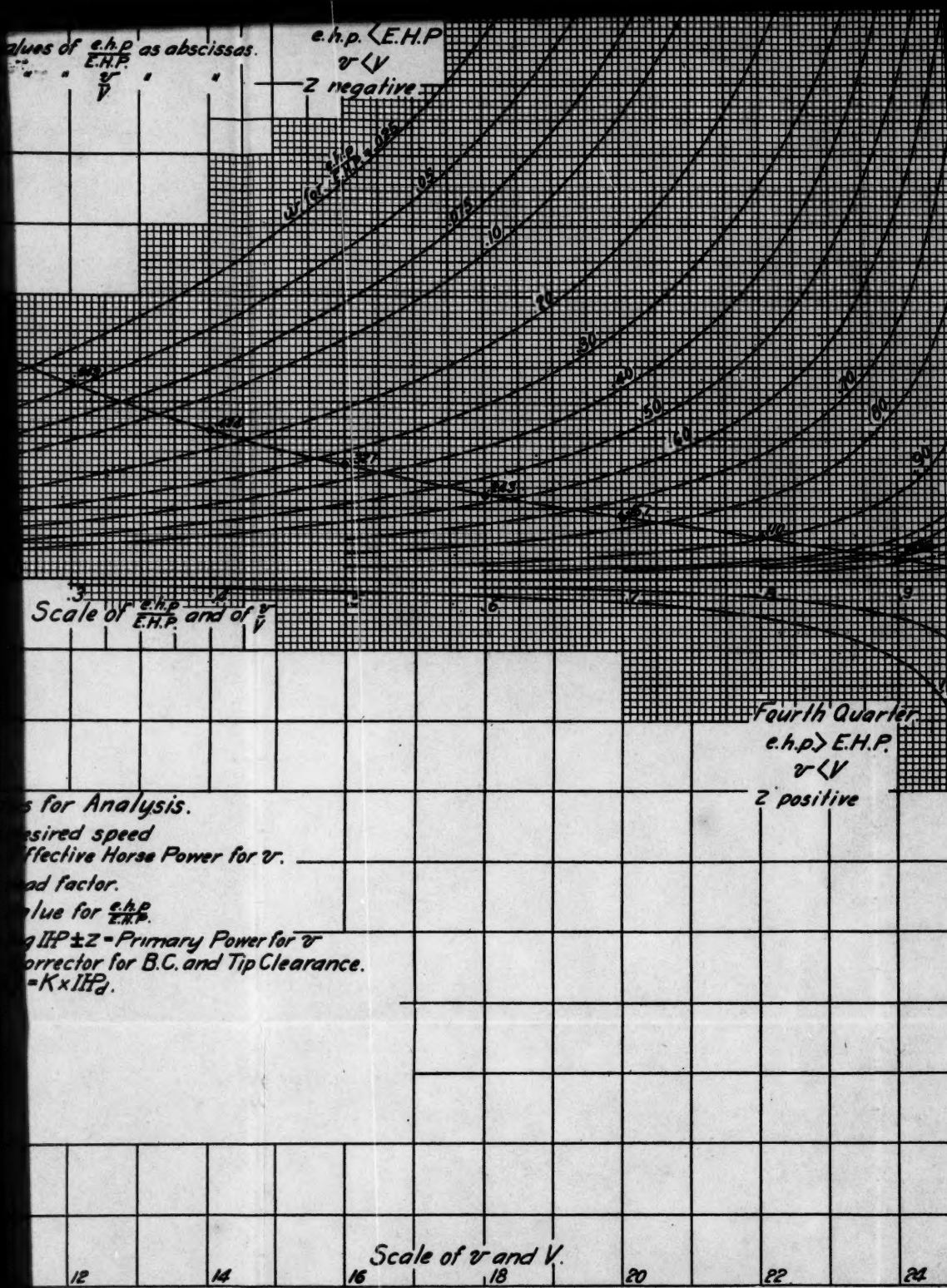
23. K = Corrector for B.C. and Tip Clearance.
24. Final $IHP_d = K \times IHP_d$.


Second Quarter.
e.h.p > E.H.P.
v > V
Z positive.
Fourth Quarter.
e.h.p > E.H.P.
v < V
Z positive
Third Quarter
e.h.p. < E.H.P.
v > V
Z negative.
of v and V.
18
20
22
24
26
28


alues of e.h.p / E.H.P. as abscissas.
v / V
e.h.p. < E.H.P
v < V
Z negative
Scale of e.h.p / E.H.P. and of v / V
Fourth Quarter.
e.h.p. > E.H.P.
v < V
Z positive
s for Analysis.
esired speed
ffective Horse Power for v.
ad factor.
lue for e.h.p / E.H.P.
g IHP ± Z = Primary Power for v
orrector for B.C. and Tip Clearance.
= K × IHPd.
Scale of v and V.
12
14
16
18
20
22
24

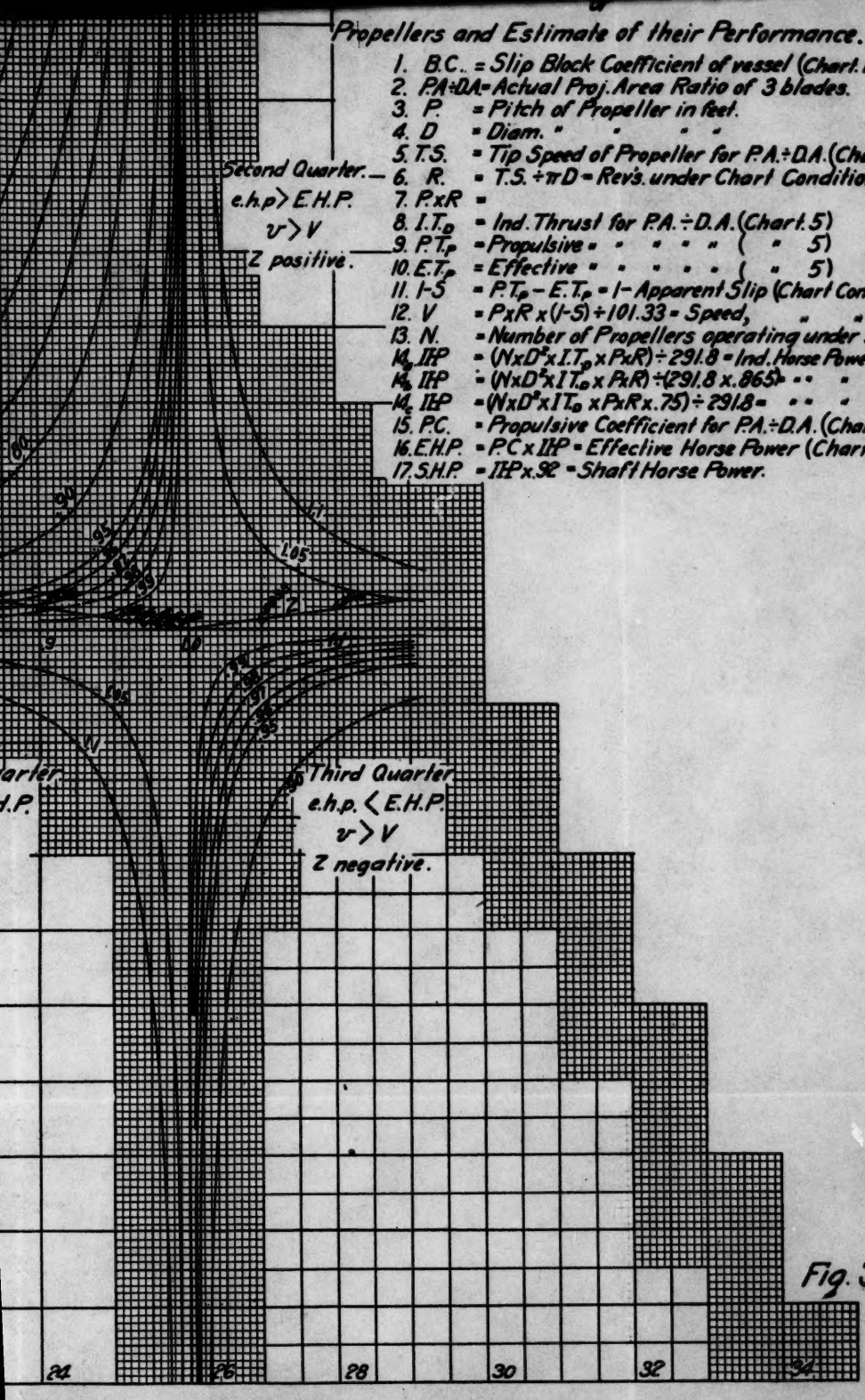

Propellers and Estimate of their Performance.
1. B.C. = Slip Block Coefficient of vessel (Chart 1)
2. P.A÷D.A. = Actual Proj. Area Ratio of 3 blades.
3. P. = Pitch of Propeller in feet.
4. D = Diam. " " "
5. T.S. = Tip Speed of Propeller for P.A.÷D.A. (Chart 5)
6. R. = T.S. ÷ πD = Rev's. under Chart Conditions.
7. P.xR =
8. I.T.p = Ind. Thrust for P.A.÷D.A. (Chart 5)
9. P.Tp = Propulsive " " " " " (" 5)
10. E.Tp = Effective " " " " " (" 5)
11. 1-S = P.Tp - E.Tp = 1- Apparent Slip (Chart Cond.)
12. V = PxR x (1-S) ÷ 101.33 = Speed, " "
13. N. = Number of Propellers operating under similar
14a IHP = (NxD²xI.Tp x PxR) ÷ 291.8 = Ind. Horse Power 3 blad
14b IHP = (NxD²xI.Tp x PxR) ÷ (291.8 x .865) = " " " 4 "
14c IHP = (NxD²xI.Tp x PxR x .75) ÷ 291.8 = " " " 2 "
15. P.C. = Propulsive Coefficient for P.A.÷D.A. (Chart 5)
16. E.H.P. = P.C x IHP = Effective Horse Power (Chart Conditi
17. S.H.P. = IHP x .98 = Shaft Horse Power.
Second Quarter.
e.h.p > E.H.P.
v > V
Z positive.
Third Quarter
e.h.p. < E.H.P.
v > V
Z negative.
24
26
28
30
32
34

Fig. 3

$$(1\text{st quad.})\ \frac{e.h.p.}{E.H.P.} < 1, \frac{v}{V} < 1.$$

$$(3\text{d quad.})\ \frac{e.h.p.}{E.H.P.} < 1, \frac{v}{V} > 1.$$

The equations, when conditions fall within either of these quadrants, take these forms (1st quad.) log $I.H.P._d$ = I.H.P. + w (log v − log V),

(3d quad.) log $I.H.P._d$ = log I.H.P − w (log v − log V).

In the 2d and 4th quadrants, which may be called the overload quadrants,

$$(2\text{d quad.})\ \frac{e.h.p.}{E.H.P.} > 1, \frac{v}{V} > 1.$$

$$(4\text{th quad.})\ \frac{e.h.p.}{E.H.P.} > 1, \frac{v}{V} < 1.$$

The equations then become for the

(2d quad.). log $I.H.P._d$ = log I.H.P. + w (log v − log V).

(4th quad.). log $I.H.P._d$ = log I.H.P. − w (log w − log V).

ESTIMATION OF REVOLUTIONS FOR CONDITIONS OTHER THAN CHART CONDITIONS.

In estimating change of revolutions caused by change of conditions, the following formula, derived from actual trial results, can be used and confidence can be placed in its accuracy:—

$$R_d = R k_1^3 \times \left(\frac{v}{V}\right)^x$$

where R = Chart Revolutions.

k_1 = Minimum value of K for slip B.C.

x = Coefficient for $\frac{v}{V}$ and $\left(\frac{e.h.p.}{E.H.P.}\right)$, Fig. 4.

The equation for diameter under these conditions becomes $D = \sqrt{(291.8 \times I.H.P.) \div (I.T._D \times P \times R)}$, where I.H.P.

= the Chart Condition of the propeller for power. Calling this diameter as affected by k_1, D_{k_1},

$$D_{k_1} = D$$
$$P_{k_1} = P$$
$$R_{k_1} = k_1^3 R$$

where D, P and R would be those necessary when $k_1 = 1$.

The equation for revolutions justifies the following statement:

RULE 5.—*If a screw propeller working in the wake of a vessel will deliver a certain effective horsepower with a certain indicated or shaft horsepower at a certain number of revolutions at a certain speed of vessel under any given condition of resistance, it will deliver the same effective horsepower, with the same indicated or shaft horsepower, while the revolutions will vary directly, as the ratio* $\left(\frac{v_2^{x_2}}{v_1^{x_1}}\right)$, *where* x_2 *and* x_1 *depend upon the ratios of the actual speeds* v_1 *and* v_2 *to the Chart Condition Speed V, at any other condition of resistance, so long as it is operating in the wake of the same hull, provided the effective thrusts do not exceed the point of cavitation.*

CAVITATION.

At one time the author advanced the statement that the point of cavitation depended upon tip speed and effective thrust. He now considers that this statement is in error and that the words *tip speed and* should be eliminated, for the tip speed at cavitation will vary with the effective horsepower being delivered while for equal effective thrusts this effective horsepower will vary inversely as the speed of ship since

$$\text{effective thrust} = \frac{\text{effective horsepower} \times 33{,}000}{\text{speed of ship} \times 101.33},$$

and a change in effective horsepower carries with it a change in revolutions. Therefore:

RULE 6.—*Should a propeller working in the wake of a certain hull cavitate at a certain speed while delivering a certain*

effective horsepower, it will, should the loading of the ship be changed, cavitate at any new speed where the effective thrust delivered is the same as that being produced under the original conditions of loading.

These cavitating thrusts will, under Chart 5, "Screw Propeller," conditions occur at about 10 per cent. increase over the effective thrusts given on that chart.

Therefore, having under chart conditions

$$E.T._p = \frac{E.H.P. \times 33{,}000}{V \times 101.33 \times 36\pi D^2},$$

the value of effective thrust per square inch of projected area at the approximate point of cavitation will be

$$E.T._{p\ cav} = \frac{1.10 \times E.H.P. \times 33{,}000}{V \times 101.33 \times 36\pi D^2}.$$

Where k_1 exceeds unity, due to decrease in density of the column of water flowing to and through the propeller, E.T.$_p$ and E.T.$_{p\ cav}$, become, respectively

$$E.T._{pk_1} = \frac{E.T._p}{k_1^3} = \frac{E.H.P. \times 33{,}000}{k_1^3 \times V \times 101.33 \times 36\pi D^2}, \text{ and}$$

$$E.T._{pk_1\ cav} = \frac{E.T._{p\ cav}}{k_1^6} = \frac{1.10 \times E.H.P. \times 33{,}000}{k_1^3 \times V \times 101.33 \times 36\pi D^2}.$$

At any other speed, v, for which an effective horsepower, *e.h.p.*, is required, the propeller will, therefore, be cavitating or on the verge of cavitation when

$$e.t._p = \frac{e.h.p. \times 33{,}000}{k_1^3 \times v \times 101.33 \times 36\pi D^2} =$$

$$E.T._{pk_1\ cav} = \frac{1.10 \times E.H.P. \times 33{,}000}{k_1^3 \times V \times 101.33 \times 36\pi D^2}$$

and this explains why with slow-speed vessels having high values of k_1 cavitation occurs at such low values of effective thrust.

POINTS TO BE CONSIDERED IN PROPELLER DESIGN AND ANALYSIS.

Before taking up the actual work of analyzing or designing propellers to illustrate the foregoing, it is necessary to fully consider and bear in mind the following points:—

(1) The effective horsepower that will be delivered by any propeller with any application of power to the propeller shaft, depends upon the actual projected area ratio of the propeller, but does not depend upon the distribution of the projected area, in other words, does not depend upon the shape of the projection.

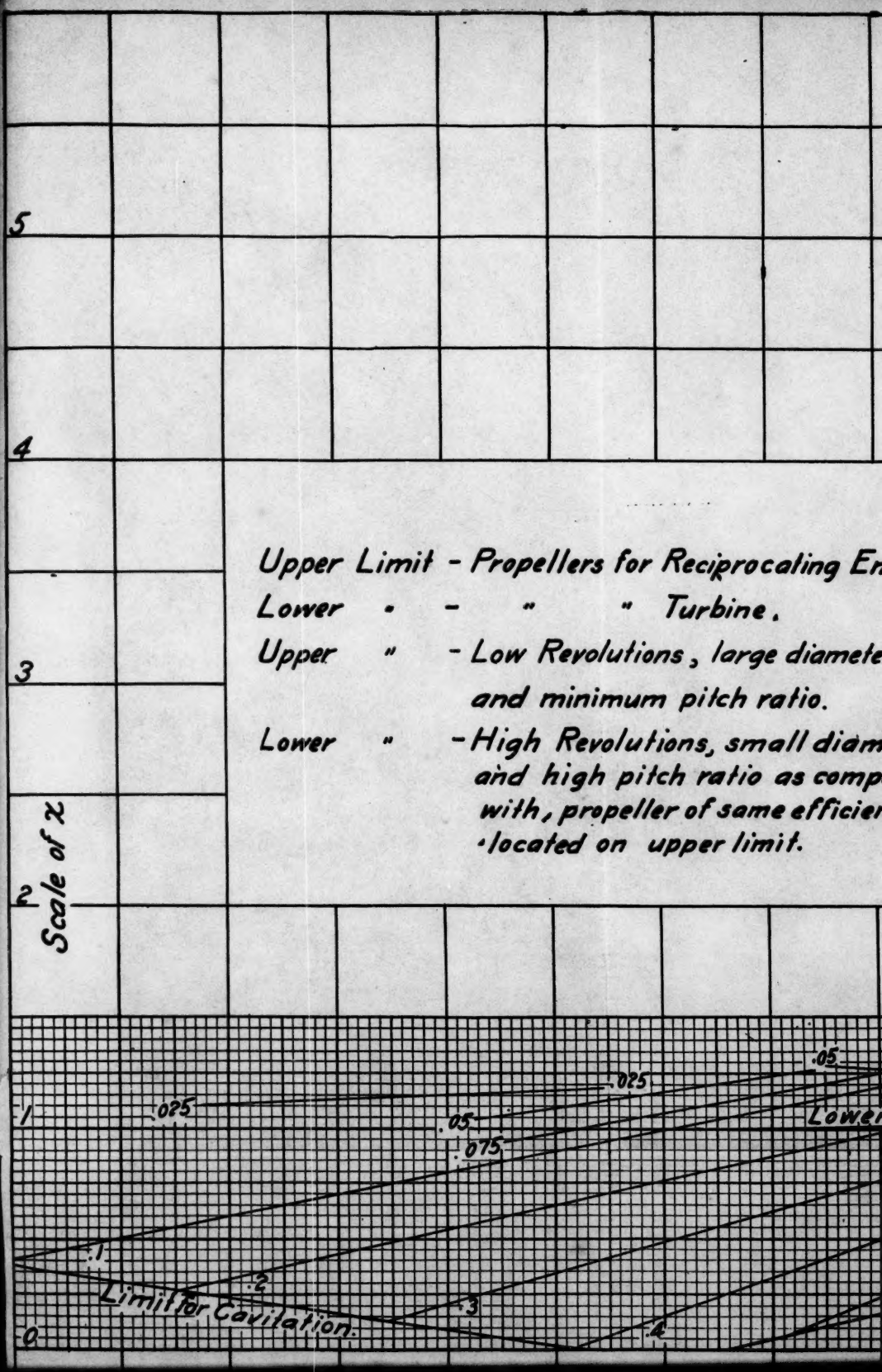
(2) Revolutions are affected by shape of projection; thus a broad-tipped blade or one with a bulbous form, will, with equal applied powers, turn slower than a blade having the standard form of projection (Chart 8, Sheet 24, "Screw Propellers"), but the thrusts will be greater, while the effect of narrowing the tip of the projection will be just the opposite.

(3) When propellers are working so close to the hull as to influence the effect of K, broad or bulbous-tipped blades will tend to increase K above that due to standard formed projections, while to narrow the tip below the standard will hold down this increase, while k_1 will apparently be affected inversely.

(4) Revolutions depend upon the rate of flow and the density of water flowing to the propeller, and are therefore affected by the value of k_1; that is, as the indicated or shaft horsepower necessary to deliver any effective horsepower increases due to an increase in k_1, the revolutions increase accordingly.

(5) The hull has no effect upon the effective horsepower that can be delivered by any given propeller except in so far as it affects the value of k_1.

(6) The hull increases or decreases the indicated or shaft horsepower, depending upon its effect upon the factor k_2.


5
4
3
2
1
0
Scale of x
Upper Limit - Propellers for Reciprocating En
Lower " - " " Turbine.
Upper " - Low Revolutions, large diamete
and minimum pitch ratio.
Lower " - High Revolutions, small diam
and high pitch ratio as comp
with, propeller of same efficie
located on upper limit.
.025
.05
.075
.1
.2
.3
.4
Limit for Cavitation.

ting Engines.

"

diameter

tio.

ll diameter

s compared

efficiency

t.

Cavitation occurs approximately where

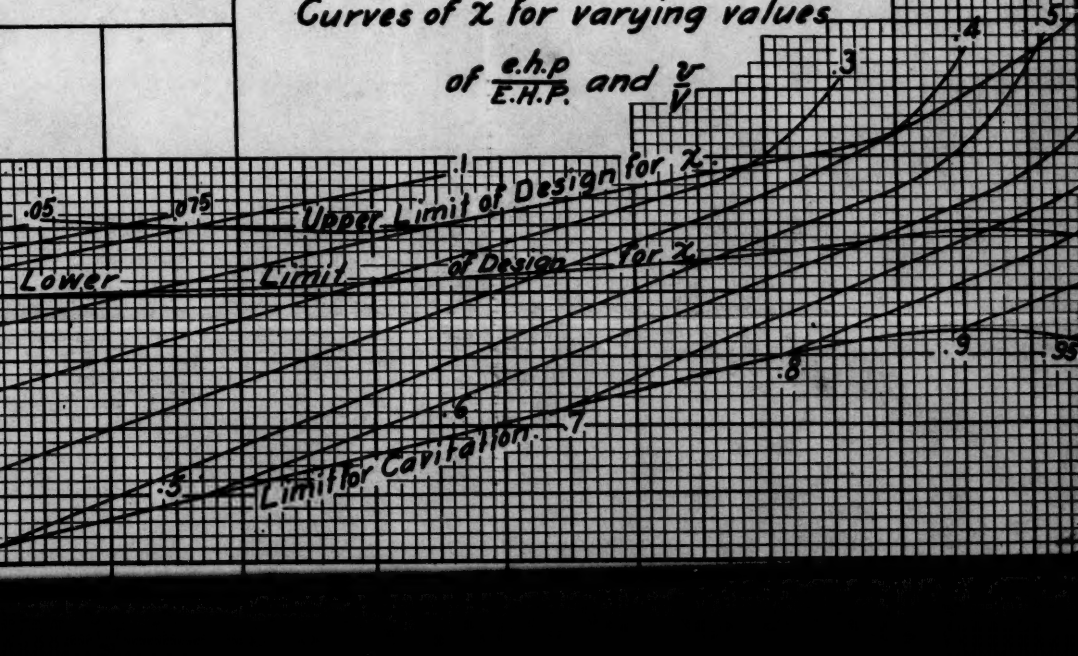
$$\frac{E.H.P. \times 33000}{\frac{P.A}{D.A} \times V \times 101.33 \times 36\pi D^2} = 1.10 \times E.T_p \text{ of Chart. 5.}$$

D being expressed in feet.

Use this Quadrant, only, for Design.
Use all Quadrants for Analysis.

First Quadrant.

Curves of χ for varying values of $\frac{e.h.p}{E.H.P.}$ and $\frac{v}{V}$



Reciprocating Engines.
Turbine. "
ns, large diameter
n pitch ratio.
ions, small diameter
ch ratio as compared
r of same efficiency
upper limit.

Cavitation occurs approximately whe

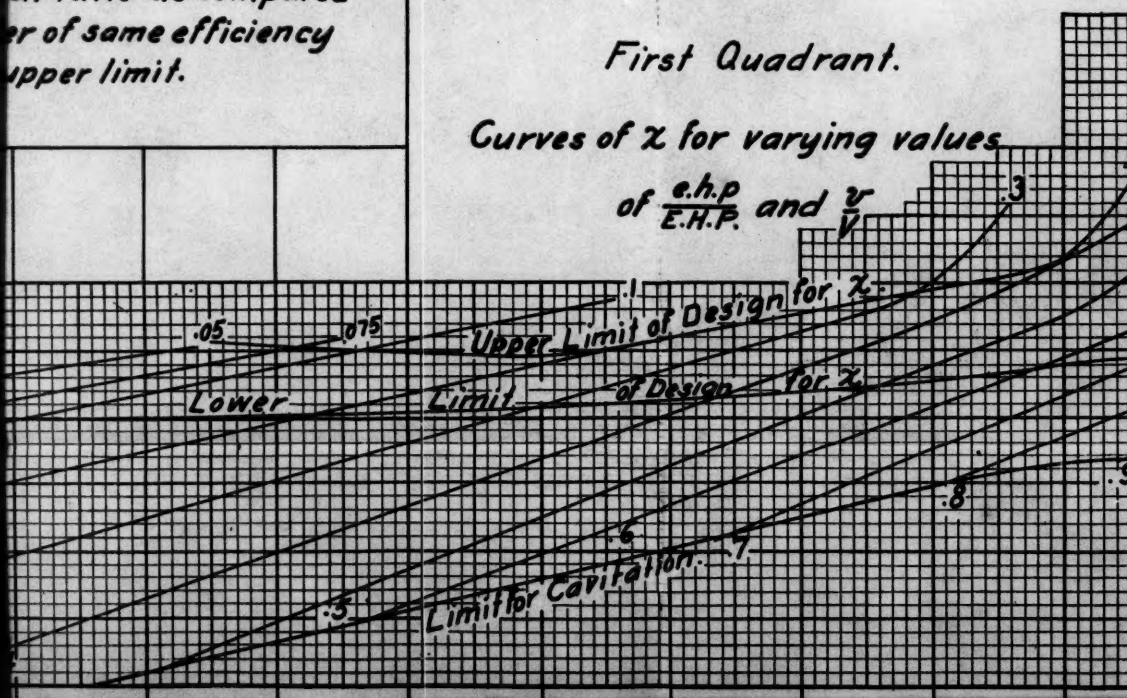
$$\frac{E.H.P. \times 33000}{\frac{P.A}{D.A} \times V \times 101.33 \times 36\pi D^2} = 1.10 \times E.T_p \text{ of Chart. 5}$$

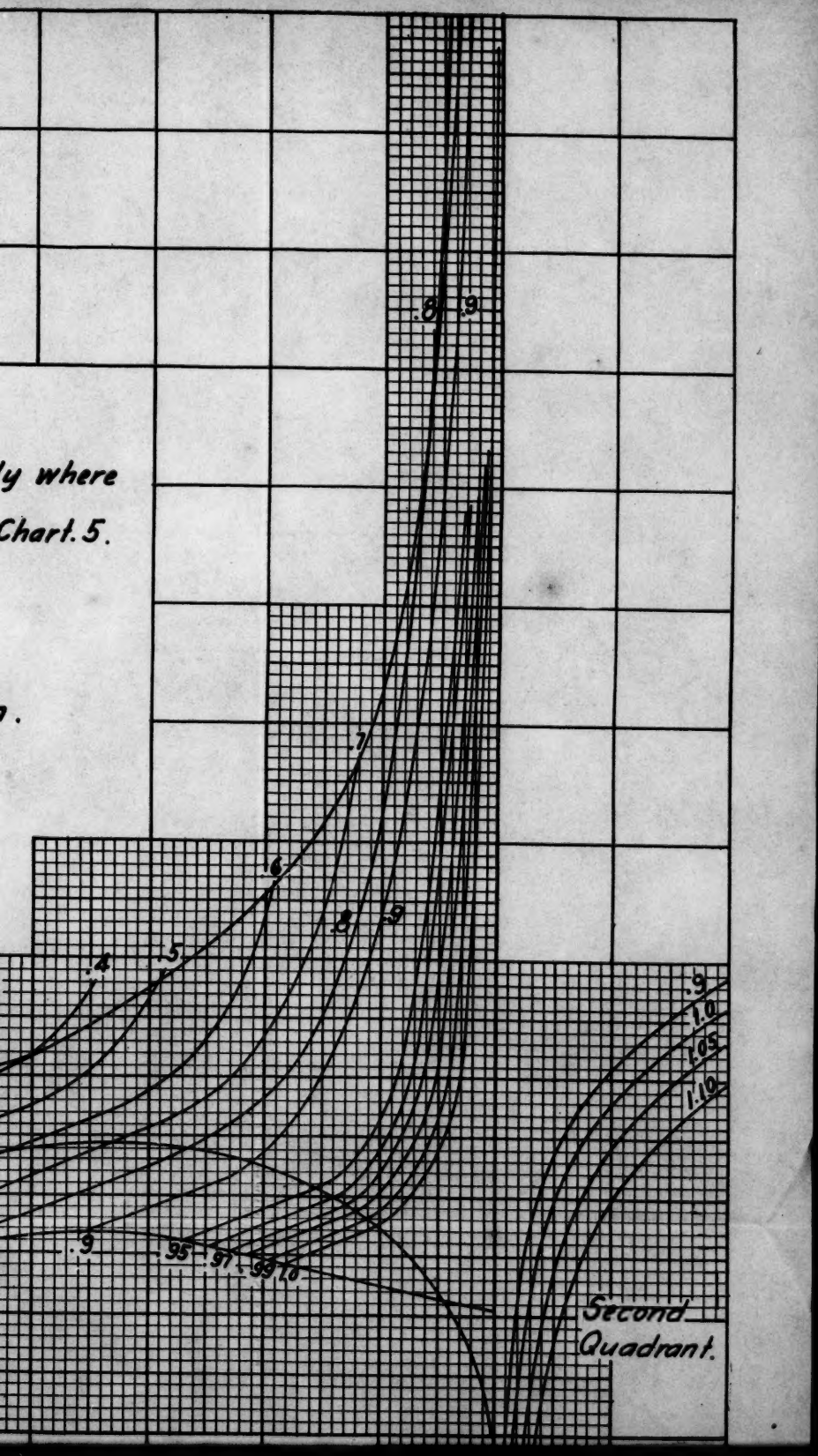
D being expressed in feet.

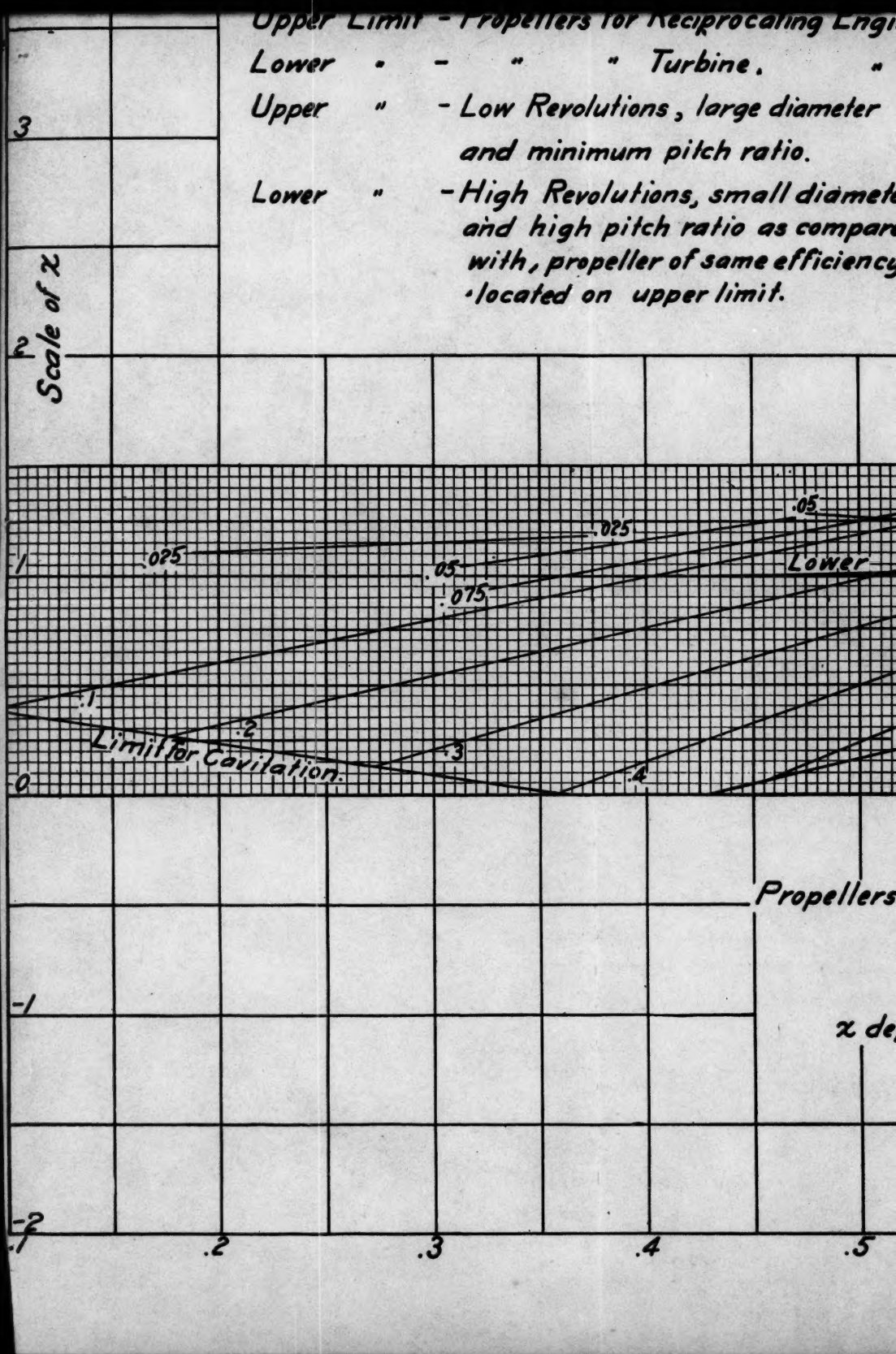
Use this Quadrant, only, for Design.
Use all Quadrants for Analysis.

First Quadrant.

Curves of χ for varying values of $\frac{e.h.p}{E.H.P.}$ and $\frac{v}{V}$




.8
.9
ly where
Chart. 5.
.7
.6
.8
.9
.4
.5
.9
1.0
1.05
1.10
.9
.95
.97
.99
1.0
Second
Quadrant.


Upper Limit - Propellers for Reciprocating Engi
Lower " - " " Turbine. "
Upper " - Low Revolutions, large diameter and minimum pitch ratio.
Lower " - High Revolutions, small diamete and high pitch ratio as compare with, propeller of same efficiency located on upper limit.
Scale of χ
3
2
1
0
-1
-2
.025
.05
.075
.1
.2
.3
.4
Limit for Cavitation.
Lower
Propellers
χ de
.1
.2
.3
.4
.5

g Engines.

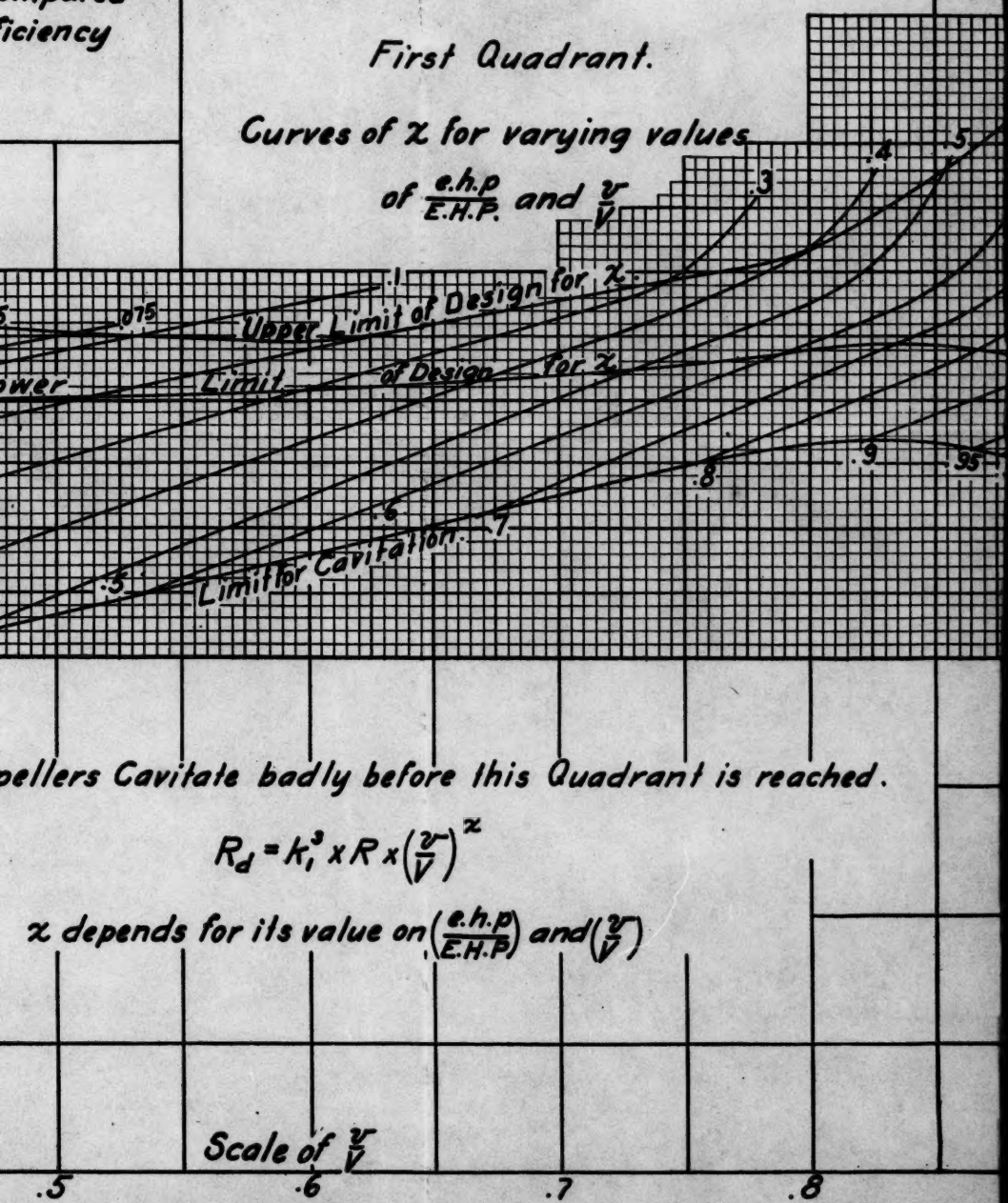
"

meter

liameter
ompared
ficiency

D being expressed in feet.

Use this Quadrant, only, for Design.
Use all Quadrants for Analysis.



pellers Cavitate badly before this Quadrant is reached.

$$R_d = k_1^3 \times R \times \left(\frac{v}{V}\right)^{\chi}$$

χ depends for its value on $\left(\frac{e.h.p}{E.H.P}\right)$ and $\left(\frac{v}{V}\right)$

Scale of $\frac{v}{V}$

.5 .6 .7 .8

Fig. 4.

... Turbine. "

...tions, large diameter

...um pitch ratio.

...lutions, small diameter

...pitch ratio as compared

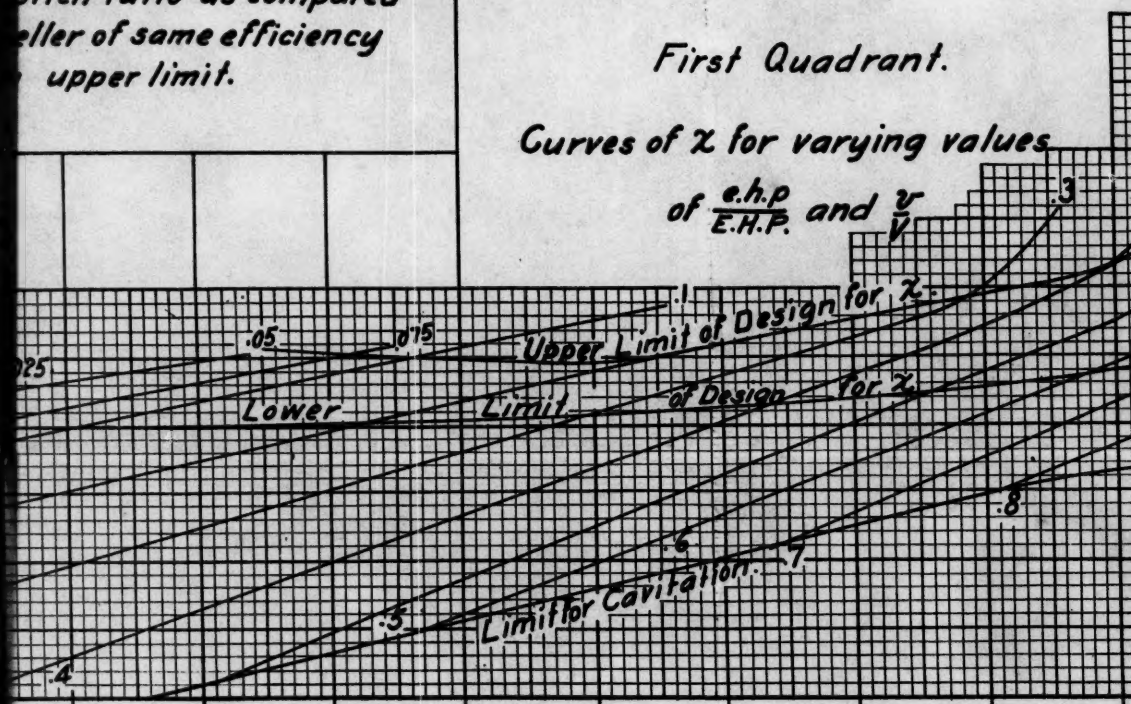
...eller of same efficiency

... upper limit.

D being expressed in feet.

Use this Quadrant, only, for Design.

Use all Quadrants for Analysis.



Propellers Cavitate badly before this Quadrant is reac

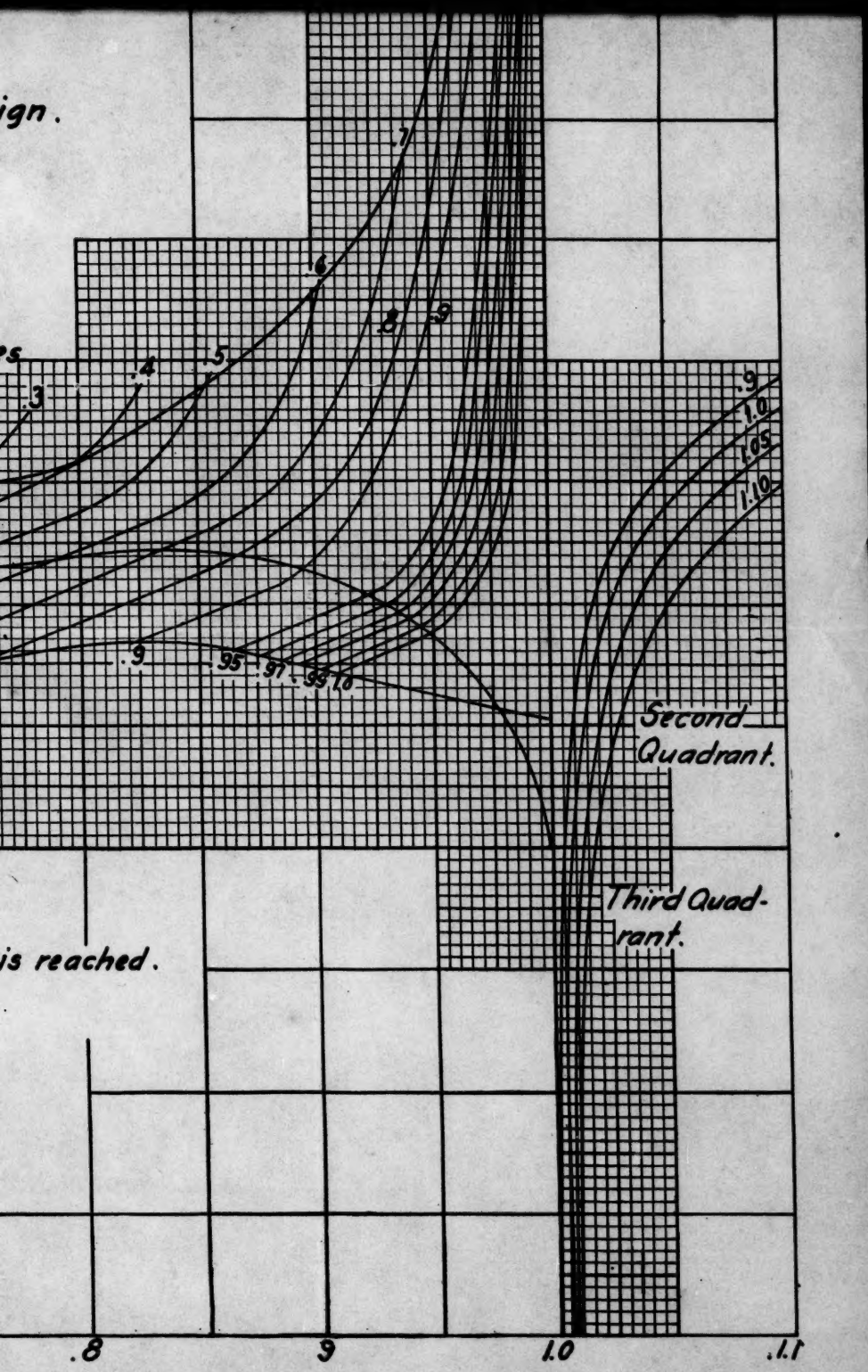
$$R_d = k_1^3 \times R \times \left(\frac{v}{V}\right)^{\chi}$$

χ depends for its value on $\left(\frac{e.h.p}{E.H.P.}\right)$ and $\left(\frac{v}{V}\right)$

Scale of $\frac{v}{V}$

.4 .5 .6 .7 .

Fig. 4.


ign.
.7
.6
.8
.9
.3
.4
.5
.9
1.0
1.05
1.10
.9
.95
.97
.99
1.0
Second Quadrant.
Third Quadrant.
is reached.
.8
.9
1.0
1.1

(7) The hull affects the revolutions, as the value of V for the Chart Condition increases with the slip block coefficient.

(8) In comparing estimate of with the actual performances the difference between model-tank conditions of hull smoothness and water conditions and those that actually exist under the trial conditions of the real ship must be appreciated. It is hardly possible to obtain an exact similarity of these conditions, and therefore the actual *e.h.p.* for the different speeds and those given by the model tank for these same speeds may differ to an appreciable degree, and a considerable difference between the estimate and the actual performance will thus be produced.

The principal causes of such differences of condition are: Roughness of ship's bottom; adverse condition of wind and sea; undue shallowness of water on trial course; poor helmsmanship; possible error in assuming that the "Law of Comparison" holds for the hull appendages.

On this latter point, however, the estimates made where trial conditions were good in all respects, agree so closely with the actual performances that it appears practically certain that

THE LAW OF COMPARISON DOES APPLY TO THE HULL APPENDAGES.

(9) Another point which must be taken into consideration is the effect of various blade sections on the efficiency of performance of the propeller.

Where different blade sections are used for several propellers having the same diameter, same form and amount of projected area, and same real (not nominal) pitch, that propeller which has blade sections offering the least resistance to motion through the water will have the greatest efficiency.

By real pitch is meant the actual pitch of the propeller as modified by the influence of the back of the blade. By nominal pitch is meant the pitch of the helical pitch surface and

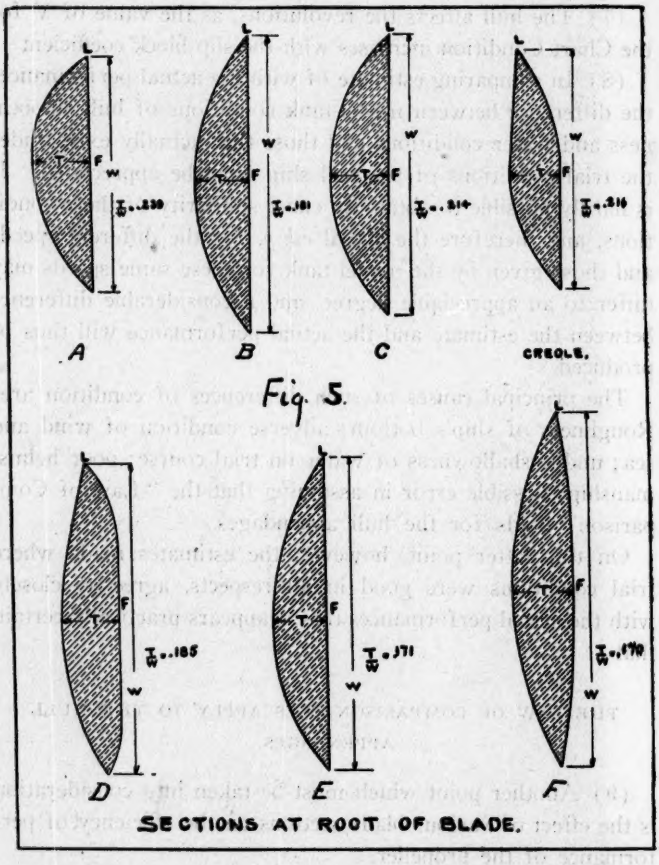


Fig. 5.

is the pitch referred to when the pitch of a propeller is spoken of.

The propellers from which the values of s were obtained were all of manganese-bronze or Monel metal; blades were all machined to pitch with thicknesses brought down to template; blade bolts, where propellers were of the built-up variety, carefully recessed into hub and fairing covering plates fitted over them; propeller blades and hubs highly polished.

The blade sections were all nearly similar to those marked B and C, Fig. 5; where differences existed, they were but slight.

Should the blades be very wide and very thin as compared with their width, variation of form of section has little effect. Should, however, the blades be narrow, the thickness ratio increases very greatly and blade section becomes of great importance. Thus a section similar to those marked Creole and D, Fig. 5, produce abnormal increases in resistance and the indicated horsepower per revolution becomes increased very considerably above that required for Sections B and C under similar conditions of hull resistance.

A modification of the section marked Creole, in which the leading edge was thrown back from the pitch face about ½ the blade thickness, and the face itself from the medium line was gradually curved back to this edge, produced an increase in k_1 of the total figure .02, such that k_1, if with the standard section equalled 1.02, with the modified section would equal $1.02 + .02 = 1.04$, and $K = (k_1 + .02 + k_2)$. This increase in k_1 is, however, not actual but is due to the blade section which causes an undue increase in real pitch, which in turn produces an increase in P × R, I.H.P., E.H.P. and V over those given by the charts.

The section marked F tends to bring the real pitch more nearly equal to the nominal pitch of the blade than does the standard sections B and C, and the blades having this section would turn up faster than blades of the same nominal pitch but of standard section, due to this one fact alone. They possibly do have slightly less resistance to motion through the water than do those of standard section, but this cannot be stated positively.

The writer only uses Section F where the material of the designed propeller is to be much weaker than the standard material, manganese-bronze. In such cases the thickness is computed both for manganese-bronze and for the weaker material, and the difference in thickness resulting is added on to the face of the blade.

ILLUSTRATIONS OF THE FOREGOING.

In illustrating the use of the curves of z, w and K, the following vessels have been taken:

1. Destroyer.—Two-shaft, reduction-gear drive. Trials run in deep water. Single strut on each shaft.
2. Destroyer.—Two-shaft, turbine-driven. Trials run in deep water. Two struts on each shaft.
3. Battleship.—Four-shaft, turbine-driven. Trials run in deep water.
4. Battleship.—Two-shaft, reciprocating engine. Sister ship to 3. Trials run in deep water.
5. Battleship.—Four-shaft, turbine-driven. Trials run in deep water.
6. Battleship.—Sister ship to 5. Different propellers. Increased value of K. Trials run in deep water.
7. Battleship.—Two-shaft, reciprocating engines. Trials run in deep water.
8. Collier.—Two-shaft, electric-driven. Trials run in deep water.
9. Gunboat.—Single-screw, reciprocating engine.

The propellers of all the above vessels are three-bladed.

In all the above cases where S.H.P. was measured by means of torsion meters, the Gary-Cummings torsion meter was used and the results obtained speak volumes in favor of the accuracy of that instrument.

In only one or two of the illustrations will the estimate of revolutions be made, as these will be sufficient to indicate the method of computation.

Case 1.—Destroyer. Slip block coefficient = .385.

P.A. ÷ D.A. = .55, 3-bladed propeller, standard form.

P = 8.636 feet.

D = 7.625.

T.S. = 11,800 (Fig. 2).

R = T.S. ÷ πD = 492.6; P × R — log 3.62880.

$I.T._D$ = 10.27 (Fig. 2).

$$I.H.P. = \frac{2 \times D^2 \times I.T._D \times P \times R}{291.8} = 17{,}410.$$

Mean T.C. very large.

P.C. = .5325 . ·. E.H.P. = 9,270, S.H.P. = 16,017.

$\frac{e.h.p.}{E.H.P.}$	e.h.p.	v	z	K	Est. $S.H.P._d$	Actual S.H.P.	$\frac{v}{V}$	x	Est. R_d.	Act'l R_d.
.025	232	10.8	1.603	1	399	400	.3263	1.18	131.4	130.5
.05	464	13.5	1.359	1	700	700	.4078	1.15	175.6	175.
.075	696	15.3	1.165	1	1,095	1,030	.4622	1.18	191.8	197.
.1	927	16.8	1.0268	1	1,506	1,510	.5075	1.21	216.8	217.
.2	1,854	20.4	.728	1	2,996	3,000	.6163	1.26	267.6	268.
.3	2,781	22.4	.5493	1	4,522	4,500	.6767	1.25	302.4	301.25
.4	3,708	23.8	.4238	1	6,037	5.900	.7190	1.25	326.2	326.25
.5	4,635	25.2	.3266	1	7,551	7,600	.7596	1.23	352.2	352.5
.6	5,562	26.3	.2432	1	9,149	9,200	.7946	1.21	374.2	372.5
.7	6,489	27.5	.1667	1	10,912	10,900	.8308	1.21	393.6	393.75
.8	7,416	28.7	.1105	1	12,419	12,650	.8671	1.18	416.3	416.25
.9	8,343	29.8	.0540	1	14,144	14,400	.9003	1.14	437.0	438.0
1.0	9,270	30.8	.0000	1	16,017	16,300	.9305	3.80	465.0	465.0

$\log R_d = \{\log R + 3 \log k_1 + x (\log v - \log V)\}$, where, in this case, $k_1 = 1.00$.

Case 2.—Number of Propellers = 2. Blades = 3.

P.A. ÷ D.A. = .607 R = T.S. ÷ πD = 600.5

P = 6′.5 P × R = log 3.59146

D = 7.5 $I.T._D$ = 12.23

T.S. = 14,150

I.H.P. = 18,410, P.C. = .5226, E.H.P. = 9,616, S.H.P. = 16,935.

$\frac{e.h.p.}{E.H.P.}$	e.h.p.	v	z	K	Est. $S.H.P._d$	Actual $S.H.P._d$
.025	240	10	1.603	1	422	440
.05	481	12.7	1.395	1	741	830
.075	723	14.5	1.165	1	1,158	1,220
.1	962	15.9	1.0268	1	1,592	1,600
.2	1,924	19.6	.7280	1	3,183	3,130
.3	2,885	21.85	.5493	1	4,781	4,750
.4	3,846	23.31	.4238	1	6,383	6,300
.5	4,808	24.55	.3266	1	7,965	7,780
.6	5,770	25.71	.2432	1	9,674	9,400
.7	6,731	26.81	.1667	1	11,537	11,260
.8	7,693	27.85	.1105	1	13,131	13,200
.9	8,654	28.82	.0540	1	14,955	14,950
1.0	9,616	29.90	.0000	1	16,935	16,750

Mean T.C. very large.

Case 3.—Slip Block Coefficient = .610 (Mean for all Propellers). Number of Propellers = 4. No. of Blades = 3. Mean T.C. Inboard Prop. = 3′.33.

P.A. ÷ D.A. = .558 — I.T.$_D$ = 10.6
P = 8′.5 — P.T.$_p$ = 9.95
D = 9.17 — E.T.$_p$ = 12.25
T.S. = 12,080 — 1 − S = .8122

$R = T.S. \div \pi D = 419.7$ — $V = \frac{P \times R \times (1 - S)}{101.33} = 28.4$

$P \times R = 3,568$

I.H.P. = 43,596, P.C. = .527, E.H.P. = 22,976, S.H.P. = 40,108. $k_1 = 1.02$.

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. S.H.P.$_d$	Actual S.H.P.$_d$	$\frac{v}{V}$	*x*	R$_d$ Est.	Act. R$_d$
.1	2,298	12.1	1.0268	1.02	3,846	3,900	.426	1.12	171.27	173.5
.2	4,595	15.05	.7280	1.02	7,653	7,800	.53	1.12	218.8	219.
.3	6,893	17.	.5493	1.02	11,549	11,600	.5986	1.14	248.13	247.5
.4	9,190	18.8	.4238	1.02	15,418	15,600	.662	1.18	273.75	273.5
.5	11,488	19.81	.3266	1.02	19,286	19,350	.6975	1.16	293.3	292.5
.6	13,786	20.51	.2438	1.02	23,368	22,900	.7222	1.12	302.3	309.0
.7	16,083	21.08	.1667	1.02	27,870	27,000	.7423	1.05	325.7	325.0
.8	18,380	21.60	.1105	1.02	31,720	31,700	.7606	.99	339.7	330.0

Case 4.—Slip B.C. = .60. Two Propellers, 3 Bladed.

P.A. ÷ D.A. = .328 — P.T.$_p$ = 8.79
P = 19′.75 — E.T.$_p$ = 10.28
D = 18′.25 — 1 − S = .8551
T.S. = 7,240 — V = 21.04
R = 126.27 — I.H.P. = 24,767
P × R = 2,494 — P.C. = .663
I.T.$_D$ = 4.35 — E.H.P. = 16,420

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. K×I.H.P.$_d$	Actual K×I.H.P.$_d$
.075	1,232	9.5	1.165	1.02	1,728	1,600
.1	1,642	10.45	1.0268	1.02	2,373	2,200
.2	3,284	13.21	.7280	1.02	4,726	4,800
.3	4,926	15.05	.5493	1.02	7,131	7,125
.4	6,568	16.45	.4238	1.02	9,521	9,300
.5	8,210	17.7	.3266	1.02	11,909	11,600
.6	9,852	18.92	.2432	1.02	14,430	14,625
.7	11,494	19.76	.1667	1.02	17,209	17,200
.8	13,136	20.33	.1105	1.02	19,587	19,600
.9	14,778	20.8	.05402	1.02	22,307	22,200
1.0	16,420	21.24	.00000	1.02	25,262	25,200
1.05	17,230	21.44	.02200	1.02	26,575	26,600
1.10	18,062	21.62	.04400	1.02	27,975	28,100

Mean T.C. = 2′.88.

Blade Section a modified "Creole" producing increase in k_1 and K from 1.00 to 1.02.

Case 5.—Slip B.C. = .62. Four Propellers, 3 Bladed. Mean T.C. of inner screws very large = 3′.5. Mean value of K for wing and inner propellers = 1.00. Slip B.C. is the mean for position of wing and inner screws.

P.A. ÷ D.A. = .501 I.T.$_D$ = 8.75
P = 8′.193 I.H.P. = 32,031
D = 9.588 P.C. = .559
T.S. = 10,680 E.H.P. = 17,905
R = 354.6 S.H.P. = 29,469
P × R = 2,905

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Estimated S.H.P.$_d$	Actual S.H.P.$_d$
.1	1,791	10.35	1.0268	1	2,771	2,750
.2	3,581	13.05	.7280	1	5,513	5,500
.3	5,372	14.85	.5493	1	8,319	8,200
.4	7,162	16.35	.4238	1	11,106	11,150
.5	8,953	17.51	.3266	1	13,892	13,900
.6	10,744	18.52	.2432	1	16,833	16,600
.7	12,534	19.41	.1667	1	20,076	19,400
.8	14,324	20.2	.1105	1	22,849	22,500
.9	16,116	20.76	.05402	1	26,022	25,700
1.0	17,905	21.24	.00000	1	29,469	29,469
1.05	18,084	21.27	.02200	1	31,000	31,000

This estimate and the actual S.H.P.$_d$ are in error, for by checking the value of k_1 by means of the actual revolutions, it is found to be 1.025, which corresponds to the minimum value of K and also the value of K for the actual mean T.C. of the vessel. The error in actual results was caused by calibrating a full length of shafting, which included an enlarged portion for a spring bearing. A short section only of this shaft was included in the torsion-meter length and this short section included the enlarged bearing.

The result was that the torsion-meter readings and the resultant powers were too small. The error also showed up in the measured steam per S.H.P., which indicated, on the basis of this power, extremely uneconomical turbines.

Case 6.—Sister ship to Case 5. 4 Propellers—3 Blades.

Mean T.C. Wing screws = 2.24. K (Wing) = 1.10.
Mean T.C. Inner screws = 2.24. K (Inner) = 1.10.
Mean K = 1.1.

P.A. ÷ D.A. = 523 P × R = 2,893
P = 8′.188 I.T.$_D$ = 9.4
D = 10′ I.H.P. = 37,192
T.S. = 11,100 P.C. = .547
R = 353.4 E.H.P. = 20,392
S.H.P. = 34,296

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	K	Estimated K × S.H.P.$_d$	Actual K × S.H.P.$_d$
.1	2,039	11.05	1.10	3,547	3,700
.2	4,078	13.9	1.10	7,057	7,100
.3	6,118	15.81	1.10	10,650	10,650
.4	8,156	17.25	1.10	14,218	14,800
.5	10,196	18.53	1.10	17,784	19,100
.6	12,236	19.61	1.10	21,549	22,600
.7	14,274	20.40	1.10	25,700	25,500
.8	16,313	21.	1.10	29,250	29,200
.9	18,353	21.46	1.10	33,313	33,500

Case 7.—Slip B.C. = .655. Two Propellers, 3 Bladed. Mean T.C. = 2′.25.

P.A. ÷ D.A. = .308 — P × R = 2,289
P = 18′ — $I.T._D$ = 4
D = 17′.25 — I.H.P. = 18,670
T.S. = 6,890 — P.C. = .672
R = 127.14 — E.H.P. = 12,546

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. K×I.H.P.d	Actual K×I.H.P.d
.1	1,255	9.85	1.0268	1.15	2,018	.....
.2	2,509	12.45	.728	1.15	4,016	.....
.3	3,764	14.02	.5493	1.15	6,061	.....
.4	5,018	15.18	.4238	1.15	8,092	7,900
.5	6,273	16.05	.3266	1.15	10,121	9,750
.6	7,528	16.85	.2432	1.15	12,264	11,950
.7	8,782	17.58	.1667	1.15	14,626	14,250
.8	10,037	18.19	.1105	1.15	16,647	16,600
.9	11,291	18.62	.05402	1.15	18,959	18,750

Case 8.—Slip B.C. = .627. Two Propellers, 3 Blades. Mean T.C. = 1′.6. K = 1.22. Vessel under Rule 3.

P.A. ÷ D.A. = .304 — $I.T._D$ = 3.85
P = 14′.436 — I.H.P. = 13,060
D = 15′.95 — P.C. = .675
T.S. = 6,740 — E.H.P. = 8,815
R = 134.4 — S.H.P. = 12,020.
P × R = 1,943

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. K×S.H.P.d	Actual K×S.H.P.d
.1	882	8.77	1.0268	1.22	1,379	1,500
.2	1,763	11.18	.728	1.22	2,743	2,780
.3	2,645	12.77	.5493	1.22	4,140	4,160
.4	3,526	14.0	.4238	1.22	5,527	5,460
.5	4,408	14.95	.3266	1.22	6,913	7,140

Case 9.—Gunboat. Single Screw. Slip B.C. = .805. Trial displacement 2 per cent. below that corresponding to model. Trials run with and against tide, one run each way for trial points. Mean T.C. = 2′.75 . ·. Minimum value of K.

P.A. ÷ D.A. = .243	$P.T._p = 7.57$
P = 10′.625	$E.T._p = 8.29$
D = 10′	1—S. = .9132
T.S. = 5,350	V = 16.31
R = 170.3	I.H.P. = 1,637
P × R = 1,809	P.C. = .694
$I.T._D = 2.64$	E.H.P. = 1,136

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. K×IHP$_d$	Act. K×IHP$_d$	$\frac{v}{V}$	x	k_1	Revs. Est.	Revs. Act.
.1	114	8.1	1.0268	1.39	214	...	.4968	1.18	1.05	...	...
.2	227	10.1	.728	1.39	426	475	.6194	1.24	1.05	108.9	109
.3	341	11.18	.5493	1.39	642	612	.6856	1.28	1.05	121.6	124
.4	454	11.99	.4238	1.39	862	760	.7353	1.31	1.05	131.8	134.9
.5	568	12.62	.3266	1.39	1,070	950	.7738	1.28	1.05	142.0	145.8

Performances of Vessels with Four-Bladed Propellers.

It is now pertinent to pass to vessels having propellers fitted with four blades and to show that these propellers check equally as well as the three-bladed ones, when certain corrections for change in efficiency have been made.

The chart condition of performance of such wheels is that for three-bladed wheels having only 3/4 the projected area ratio, the chart I.H.P. being increased over that for the three-bladed wheel by the factor 1/.865, while for a two-bladed wheel the analysis would be for a three-bladed wheel of 3/2 the actual projected area ratio, while the I.H.P. factor would become .75.

The efficiency of a propeller, however, depends upon its total projected area ratio, therefore, in the analysis of the wheel, as also in the design, the propulsive coefficient to use is the chart propulsive coefficient (Fig. 2) corresponding to the actual projected area ratio and not to the ratio of the basic three-bladed propeller.

In illustrating this point, four vessels will be given, as follows:

Case 10.—Sister ship to Case 7. Propellers of same diameter as that vessel. Twin screws.

Case 11.—Vessel of merchant type. Single screw.

Case 12.—Vessel of gunboat type. Single screw.

Case 13.—Vessel of gunboat type. Single screw.

Case 10.—Slip B.C. of vessel = .655. Tip clearance same as Case 7 . ·. K = 1.13. k_1 = 1.03, same as Case 7. P.A. ÷ D.A. (4 blades) = .391. 3 blades = .293.

P = 17'.8125 E.T.$_p$ = 9.53
D. = 17'.25 I − S = .884
T.S. = 6,525 P × R = 2,145
R = 120.4 V = 18.66
I.T.$_p$ = 3.63 I.H.P. = 18,380
P.T.$_p$ = 8.4 P.C. = .624
E.H.P. = 11,476

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. K×IHP$_d$	Act. K×IHP$_d$	$\frac{v}{V}$	*x*	k_1	Revs. Est.	Act.
.1	1,148	9.4	1.0268	1.15	1,987	2,000	.5037	1.19	1.02	56.49	54.5
.2	2,295	11.75	.728	1.15	3,954	3,850	.6154	1.23	1.02	72.31	69.
.3	3,443	13.4	.5493	1.15	5,967	5,940	.7181	1.42	1.02	79.81	80.
.4	4,590	14.6	.4238	1.15	7,966	8,080	.7824	1.50	1.02	88.39	88.4
.5	5,738	15.43	.3266	1.15	9,965	9,950	.8269	1.60	1.02	94.45	94.5
.6	6,886	16.15	.2432	1.15	12,074	11,850	.8655	1.55	1.02	102.1	100.
.7	8,033	16.83	.1667	1.15	14,400	14,200	.902	1.68	1.02	107.4	105.75
.8	9,181	17.52	.1105	1.15	16,351	16,750	.9389	2.15	1.02	111.5	111.5
.9	10,328	18.1	.05402	1.15	18,666	19,000	.970	3.4	1.02	115.2	116.

The effective horsepowers here are estimated by adding 7 per cent. to the model-tank effective horsepower curves for a displacement of 16,000 tons, the actual displacement being 17,400 tons.

Case 11.—Vessel of merchant type, 6,200 tons displacement. Slip B.C. = .66. Single Screw. Reduction Gear. Power measured forward of gear and of thrust bearing. Estimated loss through gear and bearing estimated at 4 per cent.

Exceptionally fine aft and propellers placed very low. P.A. ÷ D.A. (4 blades) = .3497. 3 blades = .2621.

P = 16′	$E.T._p = 8.92$
D = 15.5	1—S = .887
T.S. = 5,820	V = 16.76
R = 119.5	I.H.P. = 5,497
P × R = 1,912	P.C. = .65
$I.T._D = 3.02$	E.H.P = 3,573
$I.T._D = 7.92$	

Mean T.C. = 4′ ∴ K = 1.125. Shallow water (20 fathom) course.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×SHP$_d$	Act. K×SHP$_d$	$\frac{v}{V}$	x	k_1	Revs. Est.	Act.
.1	357	8.0	1.0268	1.125	581	542	.4833	1.16	1.03	56.04	55.3
.2	714	10.2	.728	1.125	1,137	1,200	.6205	1.24	1.03	72.07	71.75
.3	1,072	11.70	.5493	1.125	1,746	1,834	.713	1.35	1.03	82.5	82.5
.4	1,429	12.80	.4238	1.125	2,331	2,424	.7799	1.49	1.03	89.92	91.0
.5	1,787	13.75	.3266	1.125	2,915	3,096	.8383	1.66	1.03	97.2	98.3
.6	2,143	14.55	.2432	1.125	3,532	3,686	.8843	1.79	1.03	104.5	104.0
.7	2,501	15.3	.1667	1.125	4,213	4,124	.9274	2.27	1.03	109.9	109.0

The excess actual K × S.H.P.$_d$ over the estimate may, in this case, be caused by the effect of the shallow water in the course, as will be shown later, but this is doubtful.

Case 12.—Gunboat. Single Screw. Slip B.C. = 702. Mean T.C. = 2′.0. K =

P.A. ÷ D.A. (4 Blades) = 4. 3 Blades = 3.

P = 11′.5	$P.T._p = 8.48$
D = 9′.67	$E.T._p = 9.55$
T.S. = 6,670	1 — S = .8873
R = 219.6	V = 22.12
P × R = 2,525	I.H.P. = 3,517
$I.T_D = 3.76$	P.C. = .62
E.H.P. = 2,181	

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×IHP$_d$	Actual K×IHP$_d$	$\frac{v}{V}$	x	k_1	Revs. Est.	Act.
.1	218	9.75	1.0268	1.175	389	408	.4408	1.07	1.012	94.73	96.
.2	436	11.82	.728	1.175	747	780	.5344	1.06	1.012	117.1	117.8
.25	545	12.65	.630	1.175	970	990	.5719	.96	1.012	133.1	128.

Case 13.—Gunboat. Single Screw. Sister to Case 9. Displacement 6 per cent. below that corresponding to model. No standardization. Full-speed trial at 13.17 knots, held over outside course, one run north and one south. Result not reliable. Mean referred T.C. = 2.83 ft. P.A. ÷ D.A. (4 blades) = .338. 3 blades = .254.

P = 11′	E.T.$_p$ = 8.51
D = 9′.5	1 − S = .9131
T.S. = 5,630	V = 18.7
R = 188.6	I.H.P. = 2,114
P × R = 2,075	P.C. = .657
I.T.$_D$ = 2.85	E.H.P. = 1,389
P.T.$_p$ = 7.77	

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×IHP$_d$	Act. K×IHP$_d$	$\frac{v}{V}$	x	k_1	Revs. Est.	Act.
.1	139	8.62	1.0268	1.39	277	...	.461	1.1	1.05	93.18	...
.2	278	10.64	.728	1.39	550	...	.5691	1.13	1.05	112.9	...
.3	417	11.73	.5493	1.39	830	...	.6274	1.12	1.05	129.6	...
.4	556	12.53	.4238	1.39	1,108	...	.6702	1.08	1.05	141.7	...
.5	694	13.17	.3266	1.39	1,385	1,208	.7044	1.02	1.05	152.8	146.7

EFFECT OF INCREASE OF LOAD FOR CONSTANT SPEED ON PROPELLERS HAVING DIFFERENT DESIGN CONDITIONS.

It has often been noted in the cases of large vessels driven by turbines and by reciprocating engines, that when the resistance for any given speed was increased to any great extent, the propellers of the reciprocating engined vessel immediately

demonstrated a great superiority over those of the turbine engined one.

This superiority depends upon the ratios $\frac{e.h.p.}{E.H.P.}$ and $\frac{v}{V}$. Where $\frac{e.h.p.}{E.H.P.} = 1$, and $\frac{v}{V}$ is less than unity, the effect of load increase for any given speed is much greater than when $\frac{e.h.p.}{E.H.P.} = 1$, $\frac{v}{V} = 1$, or when $\frac{e.h.p.}{E.H.P.} = 1$, and $\frac{v}{V} > 1$. The lower the value of $\frac{v}{V}$ for $\frac{e.h.p.}{E.H.P.} = 1$, the greater will be the increase in power percentage for the given speed.

This will be illustrated by a comparison of the effect of increase in load over trial load, for any given speed for the vessels of cases 3, 4, 5 and 6 given in the foregoing.

Suppose that these vessels have just completed their preliminary acceptance trials and after being commissioned are loaded down to their full capacity with stores, fuel, ammunition and reserve feed water preparatory to a cruise during which the standard speed will be 12 knots. Required the I.H.P. or S.H.P. and revolutions for this speed under the new conditions.

The *e.h.p.'s* for 12 knots at trial condition are: Case 3, 2,250; case 4, 2,460; case 5, 2,750: case 6, 2,650.

Assuming that when fully loaded, the vessels have an increase of 10 per cent. in displacement over their designed trial conditions, the *e.h.ps.* required for these increased displacements at 12 knots become:

Case 3, 2,350; Case 4, 2,568; Case 5, 2,880; Case 6, 2,800. Analyzing for the two conditions there results as follows:

Data for Analysis.

	Case 3.	*Case 4.*	*Case 5.*	*Case 6.*
V	28.4	21.04	24.03	23.75
E.H.P	22,976	16,420	17,905	20,392
I.H.P. or S.H.P	40,108	24,767	29,469	34,296
v	12	12	12	12
e.h.p. (light)	2,250	2,460	2,750	2,650
e.h.p. (heavy)	2,350	2,568	2,880	2,800
S	.1879	.1449	.1618	.1687
$\frac{v}{V}$	.423	.57	.499	.505
$\left(\frac{e.h.p.}{E.H.P.}\right)$ (light)	.098	.1498	.153	.13
$\left(\frac{e.p.h.}{E.H.P.}\right)$ (heavy)	.103	.1560	.161	.137
z { light	1.029	.846	.837	.905
z { heavy	1.005	.829	.815	.883
K	1.02	1.02	1.025	1.10
k_1	1.02	1.02	1.025	1.025
I.H.P.$_d$ or S.H.P.$_d$ (light)	3,751	3,531	4,289	4,268
K (I.H.P.$_d$ or S.H.P.$_d$) (light)	3,827	3,601	4,396	4,695
I.H.P.$_d$ or S.H.P.$_d$ (heavy)	3,965	3,672	4,512	4,490
K (I.H.P.$_d$ or S.H.P.$_d$) (heavy)	4,044	3,745	4,625	4,939
x for $\frac{v}{V}$ and $\frac{e.h.p.}{E.H.P.}$ (light)	1.05	1.26	1.11	1.15
x for $\frac{v}{V}$ and $\frac{e.h.p.}{E.H.P.}$ (heavy)	.98	1.22	1.09	1.13
R	419.7	126.27	354.6	353.4
R_d (light)	172.5	66.0	176.1	173.3
R_d (heavy)	180.6	67.49	178.6	176.0
Per cent. increase power	5.7	4.0	5.2	5.2
Per cent. increase revolutions	4.7	2.3	1.4	1.4
$\frac{v}{V}$ for $\frac{e.h.p.}{E.H.P.} = 1$	.796	1.0095	.886	.92
(Light) actual K (I.H.P.$_d$ or S.H.P.)	3,800	3,600	4,300	4,800
(Light) actual R_d	172	66.6	176	171

It is here shown how that, immediately upon being put into cruising trim, all four vessels become more costly to drive than was shown on their trials, the reciprocating ship requiring 4 per cent. more power, while the turbine ships require an increase of 5.2, 5.2, and 5.7 per cent. increase respectively.

Now, suppose that immediately upon putting to sea they encounter such conditions of wind and sea that the effective horsepower required for each vessel to maintain the standard

speed of 12 knots, is increased by 500, then the increases over the trial conditions are:

	Case 3.	*Case 4.*	*Case 5.*	*Case 6.*
e.h.p.	2,850	3,068	3,380	3,300
e.h.p. ÷ E.H.P.	.124	.187	.189	.162
z	.925	.753	.748	.82
I.H.P.$_d$ or S.H.P.$_d$	4,767	4,374	5,265	5,191
K	1.02	1.02	1.025	1.10
k_1	1.02	1.02	1.025	1.025
K × (I.H.P.$_d$ or S.H.P.$_d$)	4,862	4,462	5,396	5,710
$\frac{v}{V}$	.423	.57	.499	.505
x	.85	1.18	1.02	1.07
R$_d$	202.0	69.05	187.9	183.2
Per ct. increase of R$_d$	17.1	4.6	6.7	5.6
Per ct. increase of K (I.H.P.$_d$ or S.H.P.$_d$)	27	24	23	22

The small propeller of Case 3 is seen to increase heavily in revolutions and power as compared with the larger propellers of the other cases.

EFFECT ON PROPULSIVE EFFICIENCY OF TRIMMING VESSELS OF CERTAIN AFTERBODY LINES DOWN BY THE STERN.

Naval Constructor D. W. Taylor, U. S. N., has demonstrated that with vessels of the ordinary battleship form, in fact, with models of Case 7, a decrease in resistance of about 1¾ per cent. occurs when the ship is trimmed approximately three feet by the stern. This percentage decrease does not constitute the only gain that occurs in such cases, for by so trimming the vessel there is a considerable increase in wake with a corresponding wake gain for the propeller, so that with the thrust deduction remaining constant, there results a large gain in propulsive efficiency, made up of the two factors, decreased resistance and wake gain. In giving examples of such cases the total gain will be treated as decrease in resistance.

Case 14.—Battleships A and B. Slip B. C. = .608. Twin screws. Vessels alike in all respects except trim. A trimmed on even keel. B trimmed 3 feet by stern. P.A. ÷ D.A. = .32 (3-bladed).

P = 19′.99 I.T.$_D$ = 4.22
D = 18′.65 I.H.P = 24,416
T.S. = 7,110 P.C. = .666
R = 121.4 E.H.P. = 16,264
P × R = 2,427

Est. z = log I.H.P. + log K − log (K × I.H.P.$_d$). Mean T.C. = 3′.19.

A								B				
$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K × I.H.P.$_d$	Actual K × I.H.P.$_d$	Actual K × I.H.P.$_d$	K	Est. z	$\frac{e.h.p.}{E.HP.}$	$e.h.p.$	$\frac{e.h.p.B}{e.h.p.A}$
.1	1,626	9.55	1.0268	1.06	2,433	2,400	.....	1.06	...	...	...	.....
.2	3,253	12.35	.728	1.06	4,841	4,900	4,200	1.06	.7897	.172	2,798	.862
.3	4,879	14.15	.5493	1.06	7,306	7,300	6,300	1.06	.6136	.258	4,196	.860
.4	6,506	15.55	.4238	1.06	9,754	9,800	8,600	1.06	.4785	.350	5,693	.875
.5	8,132	16.7	.3266	1.06	12,200	12,300	11,100	1.06	.3677	.450	7,319	.900
.6	9,758	17.75	.2432	1.06	14,784	14,900	13,500	1.06	.2827	.550	8,945	.9167
.7	11,385	18.66	.1667	1.06	17,631	17,350	16,000	1.06	.2089	.652	10,604	.9314
.8	13,011	19.45	.1105	1.06	20,067	20,000	18,500	1.06	.1458	.740	12,035	.9269
.9	14,638	20.1	.05402	1.06	22,854	22,700	20,900	1.06	.0928	.834	13,564	.9267
1.0	16,264	20.6	.00000	1.06	25,881	25,500	23,200	1.06	.0475	.898	14,605	.898
1.05	17,077	20.85	.0220	1.06	27,226	27,200	24,700	1.06	.0203	.97	15,776	.9238
1.10	17,890	21.05	.0440	1.06	28,640	28,400	26,000	1.06	.0020	.999	16,248	.9082

There is thus seen to be an average reduction in resistance combined with wake gain of 7 per cent., throwing out the three lower points where indicator errors and percentage speed errors would be a maximum, or a portion of the gain may possibly be due to a change in the value of K, the actual change occurring in the value of k_2, which has become negative and has more than eliminated the entire value of K.

Case 15.—Again, to show the results of trim with certain classes of vessels, take the case of a sister ship to Case 7. This sister was trimmed about three feet by the stern, and obtained her designed speed on about 800 I.H.P. less than the Case 7 vessel, and with three revolutions less. The propellers were alike in all respects.

Using the same hull and propeller data and the same Chart Conditions for Case 15 as for Case 7, and estimating the resistances for Case 15, they (the effective horsepowers at the

same speeds as in Case 7) are found to be as shown in the following table:

Correcting the value of K to 1.13.

v	Actual $K \times I.H.P._d$	K	Est. z	$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	$\frac{\text{Case 12}}{\text{Case 7}}$
15.18	7,300	1.13	.4609	.362	4,542	.905
16.05	9,250	1.13	.3581	460	5,771	.920
16.85	11,230	1.13	.2738	.560	7,026	.931
17.58	13,500	1.13	.1939	.670	8,406	.957
18.19	15,750	1.13	.1269	.780	9,786	.975
18.62	17,800	1.13	.0738	.870	10,915	.967

The gain is seen to increase here as the vessel passes from the low to the high speeds, as was the case in the previous vessel, the mean apparent decrease of resistance for the three lower speeds being 8.1 per cent. and for the three higher speeds 3.7 per cent., the total mean decrease being 5.85 per cent.

EFFECT OF SHOAL WATER ON THE PERFORMANCE OF PROPELLERS.

In discussing this subject, several examples will be taken; the first will be

Vessel C: Ran in deep water; in order to establish K.

2d. Vessel D: Sister ship to C, but with different propeller of slightly less diameter but with same shaft lines as C. Ran in twenty fathoms of water.

3d. Vessel E: Sister ship to Case 2. Ran in twenty fathoms of water.

4th. Vessel F: Of same type as E, but 25 feet shorter on the water line. Ran in 20 fathoms of water.

4th. Vessel G: Sister ship to Case 1 and Case 2, but propellers located differently from either Case 1, Case 2, or E. Ran in twenty fathoms of water.

With vessels running in shallow water, the effect on the

factor K is very apparent. At the low speeds K may become even less than unity and increases gradually in value until unity may become exceeded quite considerably. The effect of shallow water appears to depend, however, upon the type of vessel, heavy vessels being adversely affected through all speeds, while light vessels feel the adverse effects from 15 knots up to 26 knots. Single-screw vessels, however, appear to be unaffected up to at least 15 knots. The major part of this increase is probably due to increase of k_1, which in some cases has become so excessive as to cause very heavy vibration of the vessel. In one recent case, the side of the vessel was broken in by the water hammer caused by k_1.*

Vessel C.

Vessel C.—Slip Block Coefficient = .64.

P.A. ÷ D.A. = .315 P × R = 2,289

P = 18 feet I.T.$_D$ = 4.09

D = 17.54 feet I.H.P. = 19,647

T.S. = 6,990 feet P.C. = .67

R = 127.2 E.H.P. = 13,163

Twin Screws, 3 Blades; V = 19.65.

To find K. Speed of ships 19 knots, Revs. = 121.

$\frac{e.h.p.}{E.H.P.} = .8 \therefore e.h.p. = 10{,}530.$ K × I.H.P.$_d$ = 16,300.

v corresponding to $e.h.p.$ = 19

$$\frac{v}{V} = .9669$$

x for $\frac{v}{V}$ and $\frac{e.h.p.}{E.H.P.} = 4.25$

z for $\frac{e.h.p.}{E.H.P.} = .8 = .1105$

log I.H.P. = 4.29330

z = .11050

log I.H.P.$_d$ = 4.18280

log (K × I.H.P.$_d$) = 4.21219

log K = 0.02939 $\therefore$ K = 1.07

* It should be borne in mind that k_1 is the measure of the density of the water flowing through the propeller. As k_1 increases the density decreases.

To find k_1.

$k_1^3 R_d$ for 19 knots = 121

$\log R = 2.10449$

$x \left(\log \frac{v}{V}\right) = 9.93791$

$\log R_d = 2.04240$

$3 \log k_1 + \log R_d = 2.08279$

$3 \log k_1 = 0.04031$. ∴ . $\log k_1 = 0.01344$. ∴ .

$k_1 = 1.0314$.

Estimate of Power and Revolutions.

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	Est. I.H.P.$_d$	Est. K	Estimated K × I.H.P.$_d$	Actual K × I.H.P.$_d$	$\frac{v}{V}$	x	Est. R_d	Act. R_d
.1	1,316	10.5	1.0268	1,847	1.07	1,977	2,000	.5344	1.26	63.37	63.
.2	2,632	12.98	.728	3,675	1.07	3,933	4,275	.6606	1.32	80.74	79.
.3	3,949	14.62	.5493	5,547	1.07	5,935	6,200	.744	1.47	90.37	89.5
.4	5,265	15.6	.4238	7,405	1.07	7,923	7,600	.7939	1.55	97.6	96.2
.5	6,586	16.8	.3266	9,262	1.07	9,910	9,900	.855	1.95	102.8	104.5
.6	7,898	17.75	.2432	11,223	1.07	12,009	12,300	.9033	2.25	111.04	111.5
.7	9,214	18.45	.1667	13,385	1.07	14,322	14,450	.939	2.76	117.3	116.8
.8	10,530	19.	.1105	15,234	1.07	16,300	16,300	.9669	4.25	121.	121.

It having been shown how, in deep water, the estimated power and revolutions correspond very closely to the actual, turn now to the sister ship, D, and see the effect of the shallow course.

P.A. ÷ D.A. = .327 — I.T.$_D$ = 4.35

P = 19 feet. — V = 22.84

D = 16.5 feet. — I.H.P. = 21,540

T.S. = 7,240 — P.C. = .663

R = 139.67 — E.H.P. = 14,281

P × R = 2,654 — K = 1.07 ; k_1 = 1.0314, as before.

Twin Screws, 3 Blades.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	Est. I.H.P.$_d$	K	Est. K × I.H.P.$_d$	Actual K × I.H.P.$_d$	$\frac{v}{V}$	x	Est. R_d	Act. R_d
.1	1,428	10.75	1.0268	2,025	1.07	2,167	2,580	.4707	1.13	65.4	65.4
.2	2,856	13.3	.728	4,030	1.07	4,312	4,800	.5823	1.16	81.84	81.7
.3	4,284	14.98	.5493	6,081	1.07	6,507	6,800	.6559	1.2	92.38	92.4
.4	5,712	16.27	.4238	8,118	1.07	8,687	8,900	.7167	1.24	101.4	100.7
.5	7,140	17.28	.3266	10,154	1.07	10,865	11,100	.7566	1.21	109.35	107.5
.6	8,569	18.11	.2432	12,304	1.07	13,165	13,600	.7929	1.20	118.7	114.5
.7	9,997	18.79	.1667	14,674	1.07	15,701	16,547	.8227	1.175	121.96	120.1

By inspection it is seen that up to a speed of about 15 knots, the estimated and actual revolutions agree, while the actual power exceeds the estimated, thus indicating an increase in the value of k_2. From 15 knots upwards, the actual revolutions grow gradually lesser than the estimated, while the actual power increases above the estimated. This indicates an increase in wake gain, reducing the value k_1, accompanied at the same time by an abnormal increase in the value of k_2, due to the increased quantity and velocity of the stream-line flow from the surface of the water toward the keel, increasing the brake action on the propeller. This increase in k_2 may have been accompanied by increased hull vibration, but, due to lack of information, it cannot be stated positively whether this was the case.

The cases of vessels E, F and G in shallow water.

These vessels were all of light draught and of high speed. Vessel F being of a smaller size than the other two, her case will be examined first, and then the other two vessels, sisters of Case 2 and Case 1, deep-water tried ships, respectively.

Vessel F.

Slip B.C. = .343. Two propellers, three blades. Usual deep water values of K and k_1 = unity.

P.A. ÷ D.A. = .587 I.T.$_D$ = 11.5
P = 6.17 feet V = 29.12
D = 6.67 feet I.H.P. = 13,512
T.S. = 13,100 P.C. = .5225
R = 625.5 E.H.P. = 7,061
P × R = 3,857 S.H.P. = 12,431

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×S.H.P.$_d$	Actual K×S.H.P.$_d$	$\frac{v}{V}$	x	k_1	Est. R_d	Act. R_d
.025	177	10.	1.603	1	310	300	.3434	1.17	1	171.	182
.05	354	12.6	1.359	1	500	660	.4327	1.21	1	227.	225
.075	531	14.25	1.195	1	793	950	.4893	1.23	1	259.7	257
.1	706	15.7	1.0268	1	1,169	1,270	.5392	1.26	1	292.5	282
.2	1,412	19.4	.7280	1	2,380	2,330	.6662	1.33	1	364.4	350
.3	2,118	20.95	.5493	1	3,509	3,000	.7194	1.37	1	398.4	382
.4	2,824	23.2	.4238	1	4,685	4,500	.7967	1.57	1	437.8	429
.5	3,530	24.6	.3266	1	5,860	5,800	.8448	1.75	1	465.6	463
.6	4,236	25.9	.2432	1	7,101	7,250	.8891	1.88	1	501.5	498
.7	4,942	27.05	.1667	1	8,469	8,625	.9289	2.3	1	528.	534
.8	5,648	28.15	.1105	1	9,639	10,000	.9667	4.1	1	544.4	559
.9	6,354	29.25	.05402	1	10,973	11,400	1.0045	−3.2*	1	616.6	616
1.0	7,061	30.41	.00000	1	12,431	13,200	1.0443	1.32	1	663.9	665

* Estimated.

Discarding the lowest three speeds on account of possibly inaccurate torsion-meter readings due to the small amount of torque, it will be noted that up to 25.9 knots, excluding the 20.95 knot point where an error in plotting undoubtedly exists, the powers agree very closely while the actual revolutions rapidly decrease and then increase as compared with the estimated ones. This phenomenon indicates a heavy wake gain, reducing the value of k_1, with an equal loss due to an increase in k_2, thus maintaining a constant value of K. After passing 25.9 knots, k_1 gradually increases and then decreases again, that is, the wake at this part of the career is actually negative, while k_2 continues to hold its high value to the end of high-speed runs. k_1 here may produce heavy hull vibrations at from 27 to 29 knots speed.

Vessels E and G.

To give a better idea of conditions on these two vessels as compared with the deep-water tried vessels of Cases 1 and 2,

the following table of propeller characteristics and locations is given:

	E	Case 2.	G	Case 1.
Diameter of propeller, . . .	92.5″	90″	91.5″	92″
Pitch of propeller, ...	82″	78″	78″	104″
Projected area ratio, ...	.595	.607	.615	.55
Actual tip clearance, ...	16¾″	19″	17″	25¾″
Hub above base, ...	4′–1″	4′	3′–9½″	3′–7½″
Middle of hub, { for'd stern post, .	0′–0″	4′	7′–0″	9′–0″
Middle of hub, { abaft strut, . .	27¾″	2′	2′–0″	3′–0″

The vessels were alike in all but the following respects: The dead wood was cut away in an arch abreast the propellers of G and forward of the propellers of E and Case 2, the after end of the arch coming down slightly and forming a skag. In Case 1, while the dead wood was cut away, the arch was missing, the raised section of the keel running aft to the stern post in a perfectly straight line. Also E and Case 2 carried two supporting struts for each propeller shaft while F and Case 1 were fitted with only one.

In the order of smoothness of running they ranked in the following order: 1, Case 1; 2, Case 2; 3, E; 4, G. Case 1 vibrated only after reaching very high speeds, and then only moderately. Case 2 began vibrating at about 20 knots but was no worse than Case 1 at 29 knots. E vibrated more heavily than either 1 or 2 at high speed, while G failed, vibration commencing at about 20 knots and becoming more and more violent as the speed and power increased until the trial was brought to a finish by the side plating of the hull rupturing opposite the blade tips.

The Chart Condition and estimated performances of E and G, as compared with the actual performances will now be shown.

Vessel E.

Chart Conditions: V = 28.73; S.H.P. = 16,845; E.H.P. = 9,568; R = 557.5.

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	K	Est. K×S.H.P.$_d$	Actual K×S.H.P.$_d$	$\frac{v}{V}$	k_1	x	Est. R_d	Act. R_d.
.025	239	9.9	1	425	311	.3446	1	1.17	160.3	158
.05	476	12.	1	678	622	.4177	1	1.2	195.6	190
.075	717	14.3	1	1,075	1,100	.4977	1	1.24	234.7	231
.1	957	15.5	1	1,584	1,500	.5395	1	1.26	256.2	250
.2	1,914	19.35	1	3,151	3,000	.6735	1	1.35	327.0	314
.3	2,871	21.6	1	4,755	4,550	.7518	1	1.52	361.3	353
.4	3,828	22.9	1	6,349	5,800	.7971	1	1.58	389.6	382
.5	4,784	24.1	1	7,941	7,250	.8388	1	1.67	415.7	410
.6	5,742	25.2	1	9,622	8,900	.8771	1	1.68	447.3	440
.7	6,699	26.24	1	11,502	10,750	.9133	1	1.88	470.1	469
.8	7,656	27.25	1	13,061	12,800	.9485	1	2.5	488.5	500
.9	8,613	28.25	1	14,876	14,900	.9833	1	5.05	...	538
1.0	9,568	29.3	1	16,845	16,800	1.02	1	.985	568.4	575
1.05	10,044	30.375	1	17,720	17,400	1.057	1	1.32	600.	614

As in the case of vessel F, there is a noteworthy decrease in actual revolutions from those estimated, beginning at 15 knots. This decrease reaches a maximum at about 20 knots, as in Case F, when it begins to diminish, again disappearing at about 26 knots, thus agreeing again with Case F. Carrying out the agreement with Case F, the actual revolutions are found exceeding the estimated from the 26-knot point upward, thus indicating practically an exact reproduction of Case F conditions with Case E.

The only difference which is found between the performance characteristics of these light, fast, shallow-draught vessels and those of the heavy battleship, D, is that in the latter case the heavy loss caused by the abnormal increase in k_2 more than offsets the gain due to increased wake from 15 knots upwards, while below 15 knots, as in the cases of E and F, there is no change in wake and therefore no change in k_1, but there is quite a heavy increase in k_2.

Vessel G.

As stated already, this vessel is similar to Case 1, 2, and E, except in the small particulars previously noted, but the

propeller is quite differently located, as shown by the table. The propellers themselves, resemble very closely, those of Case 2, the differences being so slight as to preclude any possibility of their having been the cause of the vessel's failure, particularly when it is stated that the two vessels, G and Case 2, ran their trials on practically the same displacement. It appears that the cause of the failure of G can be ascribed to location principally, a possible secondary cause being the difference in the keel lines of Case 1 and G abreast of the propeller, these differences having been already given.

Should this assumption of the cause of failure be correct, the comparison of estimated and actual powers and revolutions will show the variations of k_1 or k_2, as shown in E and F, and these variations will occur within the same speed ranges, but the comparison will also show an abnormal increase in k_2 or k_1 due to the braking action of the malign downward hull and strut currents on the inboard tips of the propeller blades, or due to the decreased density of the water in the propeller race, thus producing an abnormal increase in the power without any corresponding return in thrust by this increase.

The characteristics of the propeller having been given in the table, it remains necessary to give the Chart Conditions, which are:

S.H.P. = 18,374; E.H.P. = 10,385; V = 30.51; R = 615.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×SHP$_d$	Actual K×SHP$_d$	$\frac{v}{V}$	x	k_1	Est. R$_d$	Act. R$_d$.
.1	1,039	16.25	1.0268	1	1,727	1,675	.5326	1.25	1	279.8	271.2
.2	2,078	20.25	.728	1	3,438	3,480	.6637	1.32	1	358.	343.0
.3	3,117	22.23	.5493	1	5,189	5,225	.7286	1.40	1	394.8	385.5
.4	4,154	23.72	.4238	1	6,926	7,250	.7775	1.47	1	424.8	415.0
.5	5,193	25.11	.3266	1	8,662	9,650	.823	1.52	1	457.4	467.0
.6	6,234	26.27	.2432	1	10,496	11,800	.861	1.52	1	489.8	509.0
.7	7,270	27.57	.1667	1	12,518	14,060	.9037	1.72	1	516.7	565.0
.8	8,308	28.73	.1105	1	14,247	16,200	.9417	2.25	1	537.2	602.5
.9	9,351	29.87	.5402	1	16,225	18,000	.979	4.25	1	562.1	643.0

The hull having been repaired, the strut section axes slightly altered, and the plating opposite the propeller tips heavily stiffened to withstand the water-hammer blows, the original propellers were removed and others of the following characteristics were installed:

P.A. ÷ D.A. = .603	$I.T._D = 12.05$
P = 6.67 ft.	V = 28.89
D = 7.71 ft.	I.H.P. = 18,910
T.S. = 13,900	P.C. = .5225
R = 574	E.H.P. = 9,871
P × R = 3,851	S.H.P. = 17,397

These propellers differ but slightly from those of E, being 80 inches in pitch instead of 82 inches, and having a very slightly increased projected area ratio:

The estimated and actual performances were according to the following table:

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. K×SHP$_d$	Actual K×SHP$_d$	$\frac{v}{\bar{V}}$	x	k_1	Est. R_d	Act. R_d
.1	987	15.97	1.0268	1	1,638	1,575	.5528	1.29	1	267.2	257.
.2	1,974	19.95	.728	1	3,255	3,225	.6906	1.4	1	341.8	323.5
.3	2,961	21.99	.5493	1	4,911	4,600	.7612	1.59	1	371.9	364.
.4	3,948	23.44	.4238	1	6,557	6,420	.8114	1.71	1	385.	400.
.5	4,936	24.79	.3266	1	8,201	8,675	.8581	2.01	1	422.	439.
.6	5,922	26.00	.2432	1	9,938	10,900	.9000	2.25	1	452.8	480.
.7	6,910	27.17	.1667	1	11,853	13,225	.9405	2.85	1	481.9	526.
.8	7,896	28.25	.1105	1	13,490	15,200	.9778	5.9	1	503.9	568.
.9	8,883	29.35	.5402	1	15,362	17,120	1.016	.95	1	582.7	620.
1.0	9,871	30.45	.0000	1	17.397	19,000	1.054	1.6	1	624.4	640.

The results obtained agreeing closely with those obtained with the original propellers, and there being practically no decrease in hull vibration, it became necessary to look in other directions for a panacea. As great a change in position as was possible with the existing arrangement of struts was decided upon, and after cutting about four inches off the for-

ward end of the hubs to permit of moving the propellers that amount forward on the shafts, the propellers were replaced.

The vessel was then restandardized over the measured mile and the reduction in powers over previous trials was phenomenal, while there was a very considerable reduction in the vibrations.

The estimated and actual results of these trials are shown in the following table:

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×SHP$_d$	Actual. K×SHP$_d$	$\frac{v}{V}$	x	k_1	Est. R$_d$	Act. R$_d$.
.1	987	15.97	1.0268	1	1,635	1,575	.5528	1.29	1	267.2	258
.2	1,974	19.95	.728	1	3,255	3,100	.6906	1.4	1	341.8	325.1
.3	2,961	21.99	.5493	1	4,911	4,550	.7612	1.59	1	371.9	370
.4	3,948	23.44	.4238	1	6,557	6,150	.8114	1.71	1	401.1	402
.5	4,936	24.79	.3266	1	8,201	8,150	.8581	2.01	1	422	422
.6	5,922	26.00	.2432	1	9,938	10,175	.9000	2.25	1	452.8	471
.7	6,910	27.17	.1667	1	11,853	12,225	.9405	2.85	1	481.9	506
.8	7,896	28.25	.1105	1	13,490	14,100	.9778	5.9	1	503.9	544
.9	8,883	29.35	.5402	1	15,362	16,100	1.016	.95	1	582.7	584
1.0	9,871	30.45	.0000	1	17,397	18,300	1.054	1.6	1	624.4	630

Inspection of the above results indicate that practically the same conditions exist as in the case of E, being probably slightly modified by the different position of the propeller from that of E. There is an apparent wake gain with absence of k_2 up to 26 knots. Above that speed k_1 becomes negative, causing an increase in power and revolutions. Between 28.25 and 29.35 knots the negative wake disappears and k_1 returns to its normal value of unity, the value of k_2, however, continuing to increase as the speed increases. The most striking feature of this final performance is the great reduction in power, accompanied by a reduction in hull vibrations, over the power and vibrations of the other two trials, these reductions being attributed solely to a change of a few inches in the fore-and-aft location of the propeller.

Conclusions as to the effect of shoal water on Propeller Performances.

Considerations of the above results leads to the following conclusions: Should a vessel with winged out propellers be tried over a course having approximately 20 fathoms depth of water, for speeds lower than 26 knots the revolutions will be lowered. For light-draught vessels of narrow beam as compared with length and of fine block coefficient the powers for the speeds will also be reduced. For deep-draught vessels having from medium to full block coefficient, this decrease of revolutions will be accompanied by an increase in the powers required for the given speeds.

For speeds above 26 knots, the revolutions will increase unduly until a speed of between 28 and 29 knots is reached, when the revolutions again become normal. The powers for the speeds above 26 knots will, however, increase unduly as compared with those required in deep water for the same speeds.

With vessels having single screws located directly abaft the stern post, the effect of this same depth of water on the course appears to be nil, as shown by Cases 11 and 12.

Use of Figures 1, 2 and 4 in Design.

In the author's work on "Screw Propellers" there is described a method of designing propellers to be followed when the desired revolutions and diameter of the propeller are so small that the resultant tip speed is such as correspond to the tip speed for a propeller of much greater diameter than that allowed, and working, under Chart Conditions, at a much lower number of revolutions than those desired.

Such problems are called "Problems of Reduced Diameter." By the use of Figures 1, 2 and 4, in conjunction with Chart 5, "Screw Propellers," these problems can readily be solved, and Chart 6, "Screw Propellers," can be eliminated from consideration.

On Fig. 4 are shown two cross curves marked "Limit Line

of Design for x," and the intersections of these lines with the curves of x are the starting points for the actual work of design.

As an example to illustrate the method, the following problem will be solved:

Problem in design of Propeller.

Slip block coefficient of vessel = .627.

Vessel of Rule 3 Class for determination of K. K = 1.22; $k_1 = 1.02$.

The diameter of the propeller must not exceed 16 feet 6 inches nor the revolutions about 90 per minute when running at a speed of 14 knots per hour. Find pitch and projected area ratios of the propeller and I.H.P., when the total effective horsepower to be delivered equals 3,650, divided between two propellers.

Computation.

Base ..... *e.h.p.* = ...	3,650	3,650	3,650	3,650	3,650
Assume, in order to find load point for maximum efficiency.					
(1) $\frac{e.h.p.}{E.H.P.}$ = ...	.2	.3	.4	.5	.6
(2) E.H.P. = 3,650 ÷ $\frac{e.h.p.}{E.H.P.}$	18,250	12,167	9,125	7,300	6,083
(3) R_d = ...	90	90	90	90	90
(4) k_1 = ...	1.02	1.02	1.02	1.02	1.02
(5) $\frac{v}{V}$ = lower limit line, Fig. 4	.512	.6	.672	.732	.79
(6) x for $\frac{v}{V}$	1.01	1.04	1.08	1.12	1.19
(7) $R = R_d \div \left\{ k_1^3 \times \left(\frac{v}{V}\right)^x \right\}$	166.7	144.3	130.3	120.3	112.3
(8) T.S. = $\pi \times R \times 16'.5$	8,644	7,478	6,753	6,235	5,820
(9) P.A. ÷ D.A. for T.S. (Fig. 2)	.403	.34	.298	.273	.262
(10) P.C. for P.A. ÷ D.A.	.615	.656	.678	.686	.69
(11) I.H.P. = E.H.P. ÷ P.C	29,674	18,547	13,459	10,640	8,816
(12) K = ...	1.22	1.22	1.22	1.22	1.22
(13) K × I.H.P. = (actual power) = ...	36,203	23,803	16,420	12,977	10,756
(14) z	.728	.5493	.4238	.3266	.2432
(15) K × I.H.P.$_d$ = ...	6,772	6,720	6,188	6,118	6,144

Plotting the K × I.H.P.$_d$ and $\frac{e.h.p.}{E.H.P.}$ on P.A. ÷ D.A. as abscissas, the minimum value of K × I.H.P.$_d$ is found to be:

$K \times I.H.P._d = 5{,}980$. $I.H.P. = 12{,}550$

$P.A. \div D.A. = .285$

$$\frac{e.h.p.}{E.H.P.} = .455$$

Now proceed as follows:

(16) P.A. ÷ D.A.	.285	.285	.285	.285	.285
(17) v	14	14	14	14	14
(18) I.T.$_D$ for P.A. ÷ D.A. (Fig. 2)	3.46	3.46	3.46	3.46	3.46
(19) P.T.$_p$ (Ch. 5, "Screw Propellers")	8.27	8.27	8.27	8.27	8.27
(20) E.T.$_p$ for Slip B.C. (Ch. 5)	9.49	9.59	9.49	9.49	9.49
(21) $1 - S$	.8714	.8714	.8714	.8714	.8714
(22) $\frac{v}{V}$ (Fig. 4)	.7	.75	.775	.8	.86
(23) $V = v \div \frac{v}{V}$	19.59	18.92	18.3	17.73	16.5
(24) $P \times R = (V \times 101.33) \div (1 - S)$	2,278	2,171	2,101	2,035	1,893
(25) $D = $ *$\sqrt{(291.8 \times k_1^6 \times I.H.P) \div (2 I.T._D \times P \times R)}$	16′.81	16′.57	16′.84	17′.11	17′.74
(26) x for $\frac{v}{V}$ and $\frac{e.h.p.}{E.H.P.}$ (Fig. 4)	1.09	1.28	1.36	1.48	1.75
(27) T.S. for P.A. ÷ D.A. = .285	6,340	6,340	6,340	6,340	6,340
(28) $R = T.S. \div \pi D_d$	124.7	121.8	119.8	117.9	113.7
(29) $P = (P \times R \times \pi D_d) \div T.S.$	18′.26	17′.82	17′.53	17′.27	16′.64
(30) $k_1 =$	1.02	1.02	1.02	1.02	1.02
(31) $R_d = k_1^8 R \times \left(\frac{v}{V}\right)^x$	91.81	89.4	89.9	89.94	92.7

Plotting the curves of pitch and revolutions on diameter as abscissas, the resulting propeller for 16.5 feet diameter, is found to be:

$D = 16.5$ $R = 90$

$P = 18$ feet $K \times I.H.P._D = 5{,}980$

$P.A. \div D.A. = .285$ $v = 14$ knots.

Note: All the propellers embraced in the limits of the last computations are of equal propulsive efficiency.

RELATIVE EFFICIENCIES AT VARYING LOADS.

The following table gives the relative efficiencies obtained by any projected area ratio for all conditions of load from $\frac{e.h.p.}{E.H.P.} = .1$ to $\frac{e.h.p.}{E.H.P.} = 1.1$, when delivering the same value of $e.h.p.$

*This equation is in error; k_1^8 should not appear in the equation.

$\frac{e.h.p.}{E.H.P.}$	$\frac{I.H.P._a}{I.H.P.}$	$\frac{e.h.p.}{E.H.P.}$	$\frac{I.H.P._a}{I.H.P.}$
.1	.9402	.7	.9731
.2	.9354	.8	.9692
.3	.9410	.9	.9812
.4	.9422	1.0	1.0
.5	.9428	1.05	1.0
.6	.9507	1.10	1.006

where I.H.P. is the power required to deliver *e.h.p.* under Chart Condition, which is where *e.h.p.* = E.H.P., and $I.H.P._a$ is the power required to deliver *e.h.p.* where *e.h.p.* is less than the Chart load, the values *e.h.p.* in all cases being equal.

It is seen that there is very little change in efficiency in passing from .1 load to and including .5, the maximum efficiency for all projected area ratios being at .2 load. Both the .1 and .2 load points would, however, produce very large propellers, and it is preferable, when designing for maximum efficiency, to use the .3, .4 and .5 load conditions, passing to the .6 conditions should the propellers given by the lower points prove too large in diameter for the case under consideration.

This will be the method pursued in the next problem, and this second method, may be preferable to that given in the problem just discussed.

Problem in designing for two conditions, as in cases of submarines and tug boats, or of vessels where maximum efficiency is desired at some other than the maximum speed.

To illustrate this problem, assume the case of a submarine vessel, the contract for which makes the guarantee for the submerged speed of much more importance than the guarantee of surface speed, so that it becomes necessary to design for the maximum efficiency in the submerged condition. The statement is as follows:

Slip B. C. of vessel = .35.

Guaranteed submerged speed = 10½ knots.

Total effective horsepower for this condition = 330.

Guaranteed surface speed = 14 knots.

Total effective horsepower for this condition = 530.

Required the propeller necessary to deliver the maximum possible propulsive efficiency at the submerged condition, also the revolutions and shaft horsepower for this condition.

Required the necessary shaft horsepower and revolutions for a surface speed of 14 knots.

The vessel to be fitted with twin propellers.

e.h.p. on 1 propeller = 165 at 10½ knots.

Assume, to obtain load condition for max. efficiency:

$e.h.p. \div$ E.H.P.	.3	.4	.5	.6
E.H.P. $= e.h.p. \div \frac{e.h.p.}{\text{E.H.P.}}$	550	413	330	275
Assume, P.A. $\div$ D.A.	.3	.3	.3	.3
P.C. for P.A. $\div$ D.A.	.677	.677	.677	.677
I.H.P. = E.H.P. $\div$ P.C.	812	610	487	406
z	.5493	.4238	.3266	.2432
K	1	1	1	1
$\text{I.H.P.}_d = \text{I.H.P.} \div 10^z$	229	230	230	232

As the submerged load is approximately equal to .6 of the surface load, and as it is desirable to keep the diameter as low as possible, use the .5 condition first, as it will also give a slight gain at the surface.

I.H.P.	487.	487.	487.	487.
P.A. $\div$ D.A	.3	.3	.3	.3
I.T._D for P.A. $\div$ D.A.	3.77	3.77	3.77	3.77
P.T._P	8.49	8.49	8.49	8.49
E.T._P	10.31	10.31	10.31	10.31
1 − S	.8235	.8235	.8235	.8235
v	10.5	10.5	10.5	10.5
$\frac{v}{V}$ (Fig. 4) for $\frac{e.h.p.}{\text{E.H.P.}} = .5$	.732	.78	.82	.855
V	14.35	13.46	12.81	12.28
P × R = (V × 101.33) $\div$ (1 − S)	1,765	1,657	1,576	1,511
$k_1 = 1$	1	1	1	1
$D = \sqrt{(291.8 \times \text{I.H.P.}) \div (\text{I.T.}_p \times P \times R)} =$	4′.621	4′.770	4′.891	4′.995
T.S. for P.A. $\div$ D.A.	6,670	6,670	6,670	6,670
P	3′.841	3′.640	3′.630	3′.555
R	459.3	444.8	434.1	425.1
x for $\frac{v}{V}$ and $\frac{e.h.p.}{\text{E.H.P.}} = .5 =$	1.125	1.3	1.5	1.92
$R_d = R \times \left(\frac{v}{V}\right)^x$	323.5	322.	322.3	314.7
$\text{S.H.P.}_d = 230 \times .92$	212.	212.	212.	212.

Should these revolutions be too high, in order to go to lower revolutions it would be necessary to try again, using a load factor $\frac{e.h.p.}{E.H.P.} = .1$ or .2, depending on the amount of reduction desired. This could still further be decreased by decreasing the value of P.A. ÷ D.A. to say .25.

Assuming, though, for sake of illustration, that the revolutions obtained in the foregoing work were satisfactory, to obtain the surface condition proceed thus, using the load condition and propeller where V is greater than 14, namely,

$$v = 14$$

$$V = 14.35$$

$$\frac{v}{V} = .975$$

$$\frac{e.h.p.}{E.H.P.} = \frac{530}{550} = .964$$

x for $\frac{v}{V}$ and $\frac{e.h.p.}{E.H.P.} = 2.15$

z for $\frac{e.h.p.}{E.H.P.} = .964 = .022$

$$S.H.P._d = (487 \div 10^z) \times .92 = 426$$

$$R_d = R \times \left(\frac{v}{V}\right)^x = 459.3 \times \overline{.975}^{2.15} = 435.$$

In using Fig. 4, it is necessary to remember the following:

1. With constant value of $\frac{e.h.p.}{E.H.P.}$ and of $\frac{P.A.}{D.A.}$, the diameter of the propeller will decrease, the pitch and revolution increase as $\frac{v}{V}$ decreases.

2. With constant value of P.A. ÷ D.A., the revolutions will decrease while the diameter and the pitch will increase as the value $\frac{e.h.p.}{E.H.P.}$ decreases.

3. With constant values of *e.h.p.* ÷ E.H.P. and of v ÷ V, the pitch and diameter of the resultant propellers will

decrease as the projected area ratio is increased, while the revolutions will rapidly increase.

CONCLUSION.

The foregoing work is submitted as at least a partial solution of the "Mystery of the Screw Propeller."

It shows—

1. That the efficiency of performance of the propeller is seriously affected by its position in relation to the hull of the vessel which it is driving, and by the fullness of the hull lines at the after body.

It shows—

2. That for any given propeller working under constant hull conditions of form, the effective horsepower for any given engine power remains constant, and this is independent of the speed of the vessel.

It shows—

3. That for any given propeller working under constant hull conditions of form, the revolutions necessary to deliver any given effective horsepower vary with the speed of ship, the engine power remaining constant.

It shows—

4. That model tank effective horsepower curves are correct, and that the appendage resistance varies according to the "laws of comparison."

It shows—

5. That trials of vessels over shallow water courses should be prohibited, as such courses change completely the character of the water flow to the propeller.

It shows—

6. That deep-water trials, by the close agreement of actual with estimated results, demonstrate the correctness of the above statements.

It shows—

7. The great accuracy of torsion meters and indicators used on official trials, and that the large estimate of from three to

five per cent. error usually credited to these instruments is incorrect, particularly at high powers.

It shows—

8. That results obtained from model-tank experiments with model propellers will be correct when the model is properly proportioned to the full-size propeller, *which does not mean that the pitch ratio of the propeller and of the model shall be the same,* and the Chart Condition of the model propeller, working with no hull, or rather behind a phantom ship, can be calculated.

It shows, finally—

9. *That where vessels are entered into competition based upon relative efficiencies of their propelling machinery, it is a positive error to base these efficiencies upon speed or revolutions, as both speed and revolutions for equal load upon the propellers, represented by equal effective horsepowers, vary with the loading of the hull, foulness of ships' bottoms, weather and sea, while the engine power for this effective horsepower remains constant, except in cases where propellers are foul or their blades distorted. It shows, therefore, that the true criterion for efficiency of propelling machinery should be indicated or shaft horsepower developed by that machinery, and that the speed of ship and revolutions of propellers should be neglected.*

DESCRIPTION OF MAIN PROPELLING MACHINERY FOR THE U. S. S. *MAUMEE.*

By Lieutenant C. W. Nimitz, U. S. N., Member.

The U. S. S. *Maumee* is one of two twin-screw fuel ships authorized by the Naval Appropriation Act of August 22, 1912, "to cost, exclusive of armor and armament, not to exceed one million one hundred and forty thousand dollars each, and which shall be built in navy yards, one to be built in a navy yard on the Pacific coast." These two vessels, the *Maumee* and her sister ship, the *Kanawha,* were built in the Mare Island Navy Yard to carry fuel and minor supplies for the fleet.

The *Kanawha,* now in commission with a merchant complement, is propelled by two triple-expansion steam engines of 2,600 I.H.P. each, while the *Maumee* is to have for propelling machinery two Diesel engines of 2,500 B.H.P. each. At the time these vessels were authorized there were already afloat a number of motor ships whose performance warranted the adoption in our service of Diesel motors of the heavy, slow-speed type for our auxiliary vessels.

The sum of money available for the construction of the *Maumee* was based on steam-engine propulsion and was not sufficient to cover the entire cost of experimental and construction work of a pair of large Diesel engines.

In order to introduce this type of motive power into our service, a Naval Appropriation Act of March 4, 1913, provided "that the un-obligated balances under the appropriation 'Steam Machinery' for the fiscal years ending June thirtieth, nineteen hundred and twelve, and June thirtieth, nineteen hundred and thirteen, not exceeding $250,000, are hereby reappropriated and made available for the development of a type of heavy-oil engine suitable for use in one of the fuel ships

authorized by the Act approved August twenty-second, nineteen hundred and twelve, and the expenditure thus incurred shall not be a charge against the limit of cost of such vessel."

It is the present intention of the Navy Department to man the *Maumee* with officers of the Navy and an enlisted crew.

The two ships are similar except as to their propelling machinery, which, however, will be kept within the same machinery spaces on both vessels, that is, between Frames 136 and 166. Externally their appearance will be the same, except that the smokepipe of the *Maumee* will come out of the after end of the machinery space while that on the *Kanawha* will come out of the forward part of the machinery space. The *Kanawha* has four Babcock & Wilcox boilers forward of the main engines, while the *Maumee* will have two boilers of the same make but smaller in size just abaft the main engines.

These two vessels will afford an excellent opportunity of comparing the two types of machinery, as they are not only sister ships, but they will be employed on identically the same duty and will probably burn the same grade of fuel. It would be premature at this writing to attempt any comparison between the two vessels, except that the *Maumee's* propelling plant will be more costly and considerably heavier than that of the *Kanawha.* A comparison of weights, however, should also include the fuel to cruise considerable distances, in which latter case the question of weight will be in favor of the *Maumee's* plant, owing to the unquestioned higher fuel economy of the Diesel motor over that of the steam engine.

The principal characteristics of these two vessels are given as follows:

Length between perpendiculars, feet	455
Beam, feet	56
Draught (loaded), feet and inches	26–4
Tonnage (loaded), about, tons	15,000
Speed (loaded), knots	14.5
Cargo capacity, fuel oil, tons	10,000

The main propelling machinery of the *Maumee* is being built at the Navy Yard, New York, and consists of two single-acting, two-cycle, six-cylinder, Diesel engines of the slow-speed, cross-head, heavy-duty type. Each engine will develop its full power of 2,500 B.H.P. at 130 r.p.m.

In order to gain a maximum of Diesel engine experience in a minimum of time, the Bureau of Steam Engineering decided to obtain the manufacturing rights and a set of working plans from some company already engaged in that line of work, and to build the engines in one of the navy yards. The Navy Department, therefore, obtained bids from various shipbuilding and manufacturing companies, both at home and abroad, and finally contracted with the Electric Boat Company, of Groton, Connecticut, American licensees of the Maschinenfabrik-Augsburg-Nürnberg, or M. A. N. Co. of Nuremberg, Germany.

The Electric Boat Company, through a subsidiary company, the New London Ship and Engine Co., manufactures two-cycle, single-acting, Diesel engines of the M. A. N. type for submarines and other marine work. This company also manufactures a four-cycle Diesel engine for marine use, but up to date has not constructed engines of the heavy-duty, slow-speed type used in ocean-going vessels.

The contractor obtained the original design and drawings from the home company, translated all figures from the metric to the English system, and supplied the Government with a set of retraced plans, whereon only slight departures from the original design were made. The Navy Yard, New York, checked the translated drawings by comparison with the original German drawings and, after approval by the Bureau of Steam Engineering, built the engines, following the drawings as closely as conditions would permit. It will not be out of place here to observe that it would probably have been cheaper to build the engines using the original metric figures and scales. All the translated figures contained decimals to the thousandth part of an inch, which made the checking ex-

tremely tedious and the work in the shop, where English scales were used, slower than it would have been otherwise. The decimals made it difficult for the man at the machine tool to distinguish between important and unimportant dimensions, so that the result showed considerable time spent in making all dimensions correct.

All parts of the engines were made in the navy yard, except the very few steel castings required, and the heavy connecting rod, piston rod and thrust and line-shaft forgings. These forgings were purchased rough turned and finished in the yard. The crank shaft was purchased finished from the Erie Forge Company of Erie, Pa.

Each main engine, turning outboard going ahead, drives the following attached auxiliaries:

Three high-pressure air compressors for fuel injection and for charging air-starting bottles.

Three scavenger compressors for cylinder scavenging.

Two salt-water pumps for cooling all water jacketed parts of engine except pistons.

One fresh-water pump for piston cooling.

One lubricating-oil pump for thrust block, main, crank pin, and crosshead bearing lubrication.

Two general service water pumps for fire, bilge or sanitary purposes.

Twelve mechanical lubricators for cylinders and for minor bearing lubrication.

One fuel-oil supply pump for pumping fuel from ship's bunker tank to engine supply tanks.

Six fuel-oil measuring pumps for supplying fuel to cylinders from engine-room tanks.

One speed governor.

One tachometer.

One revolution counter.

Referring now to the general arrangement plans of the engines, it will be seen that the scavenging pumps are mounted on the outboard columns of the even-numbered cylinders from

forward, and that they are driven by means of links and beams from the main crossheads of the engine. Each scavenger compressor is double acting and draws the air for both sides of the pistons through the common opening on the outboard side of the scavenger-cylinder casting (see general plan, "section through scavenger pump,") and then through twelve flat-disc suction valves, six for the top and six for the bottom of the compressor. The air, after compression to about eight pounds pressure, passes, via twelve discharge valves similar to the suction valves, through two coolers to the scavenger receiver, from which its admission to the cylinders is controlled by the cam shaft. The suction and discharge valves are assembled in six units, each unit consisting of two suction and two discharge valves mounted on one stem. By removing the valve bonnets the valve units are readily accessible and easily removed. The scavenger cylinder is lubricated at about its middle by four supply pipes, equally spaced, from the mechanical lubricators.

Directly under each scavenger compressor and driven by the same crosshead that drives the scavenger, are two pumps, which are used as follows:

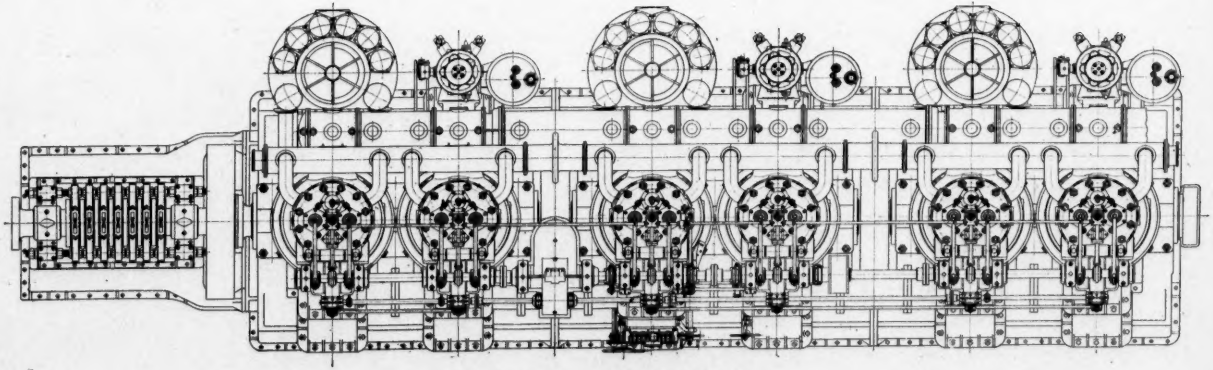
Under No. 1 scavenger compressor, forward pump for fresh water for cooling pistons; after pump for lubrication of main, crank pin, crosshead, and thrust block bearings.

Under No. 2 scavenger compressor, forward pump for salt water for cooling all parts of engine, except pistons; after pump, same as for forward pump.

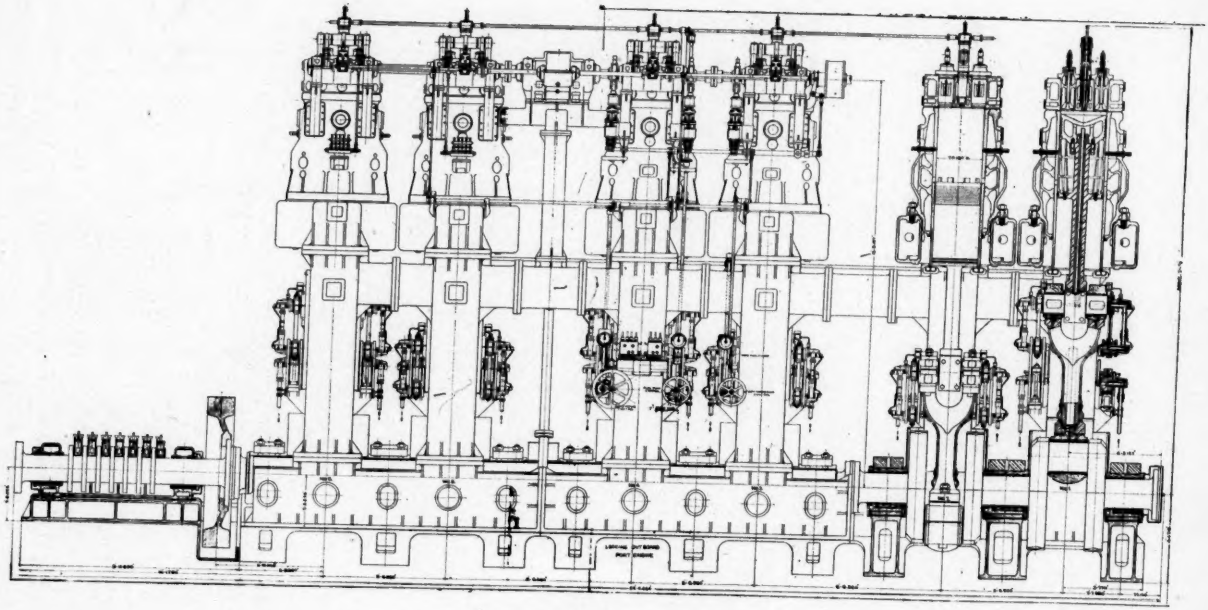
Under No. 3 scavenger compressor, forward pump for bilge and sanitary system; after pump for bilge and sanitary system.

All of the pumps under the scavenger compressors are of the same size and have their valve chambers attached to their forward and after sides respectively.

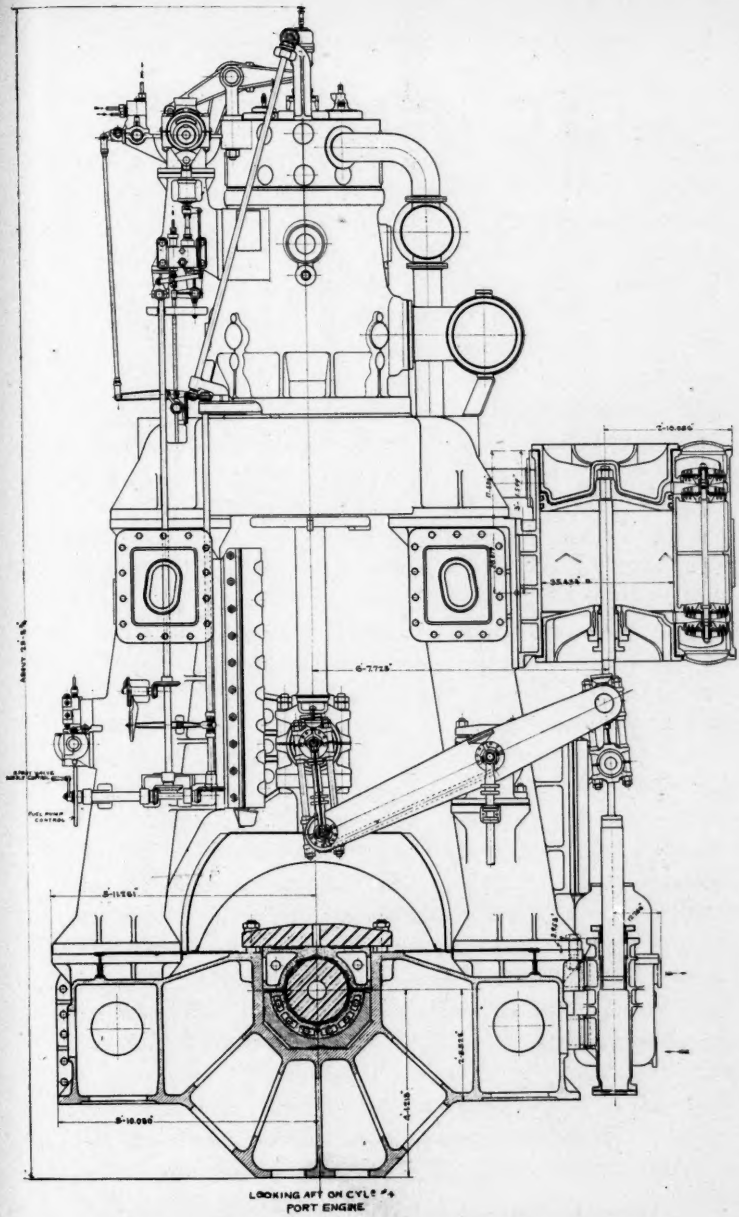
The high-pressure air compressors are mounted on the outboard columns of the odd-numbered cylinders and are driven off the main crossheads by links and beams. As in the case of the scavenger compressors, the beam at its outboard end



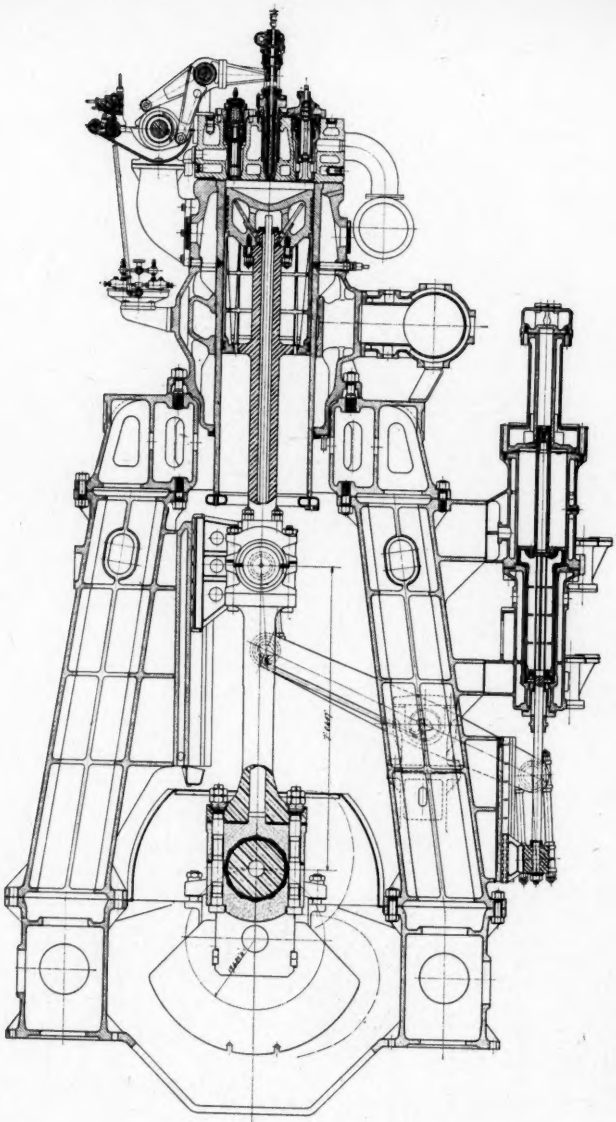
GENERAL ARRANGEMENT PLAN.



GENERAL ARRANGEMENT.



SECTION THROUGH SCAVENGER PUMP.



SECTIONAL ARRANGEMENT THROUGH WORKING CYLINDERS.

drives a smaller crosshead traveling in a guide back of the column. Each compressor is of the three-stage, tandem-piston type. The low-pressure stage is double-acting, while the middle and high stages are single-acting. Each compressor has on its forward side an intercooler, which contains the cooling coils for all three stages. The piston and discharge valves of the low stage are of the flat-disc type, while those of the higher stages are of the poppet type. The three compressors deliver to a common pipe which conveys the air through separators to the spray-air bottle, from whence it leads to the fuel-valve bodies in the cylinder heads. The spray-air bottle has on it an overflow valve, whereby air in excess of the necessary spray air is passed into the bottles for storing the starting air. Two of the attached compressors are sufficient to supply injection air at full power while the third compressor acts as a reserve. The reserve compressor runs idle when the engine is running, although it can be easily disconnected from its crosshead. The low-pressure suction valves of all the high-pressure compressors are controlled from the handling platform of the engine as well as at each compressor. Any desired pressure up to the maximum can be maintained at will.

A small fuel-oil supply pump is attached to the after side of No. 1 outboard column and is driven by the beam. This pump maintains the fuel supply from the ship's bunkers to the engine-room supply tanks. The fuel-oil measuring or cylinder-supply pumps are six in number, one for each cylinder. Two pumps are contained in one body and all the pumps are driven by eccentrics from the main cam shaft. Fuel pumps for cylinders 1 and 2 are located on the after side of after cam-shaft bracket on No. 3 cylinder. These two pumps are in one heavy forging, both pistons being connected to the same crosshead. Fuel pumps for the remainder of the cylinders are similarly driven and are located as follows:

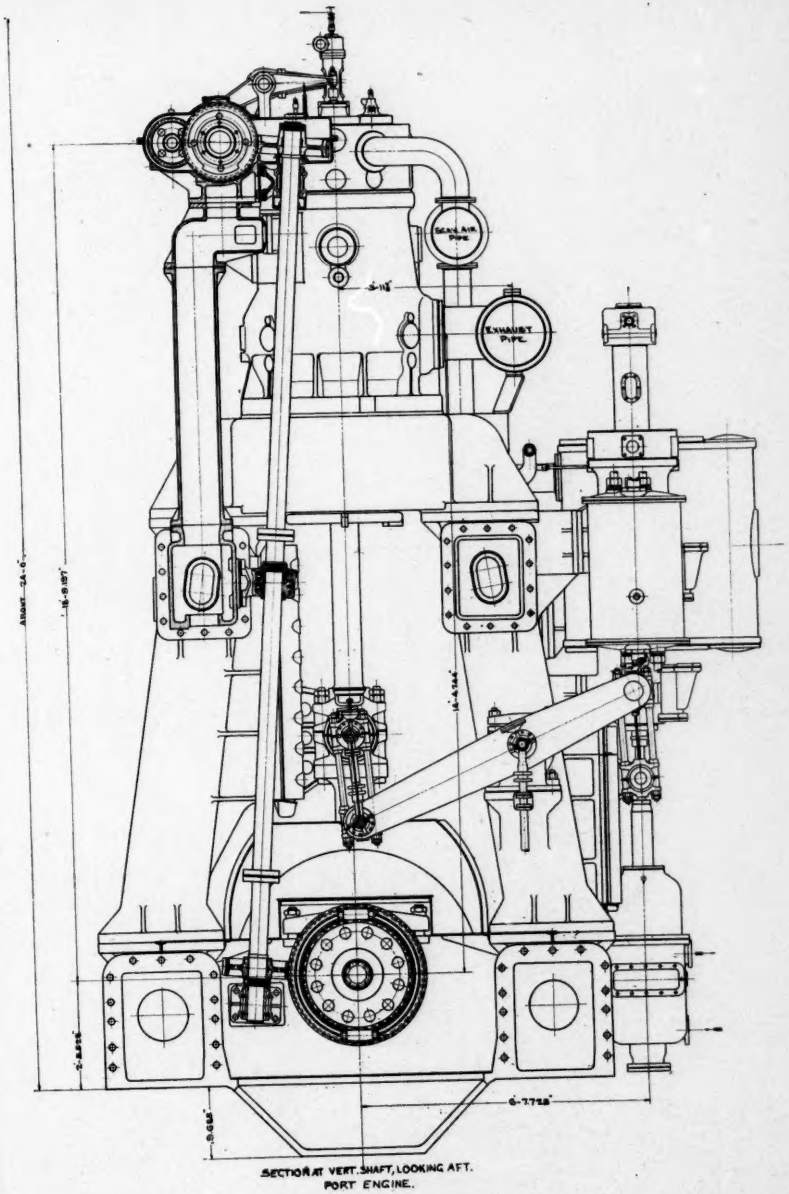
For cylinders 3 and 4, on forward side of forward cam-shaft bracket on No. 4 cylinder.

For cylinders 5 and 6, on after side of after cam-shaft bracket on No. 4 cylinder.

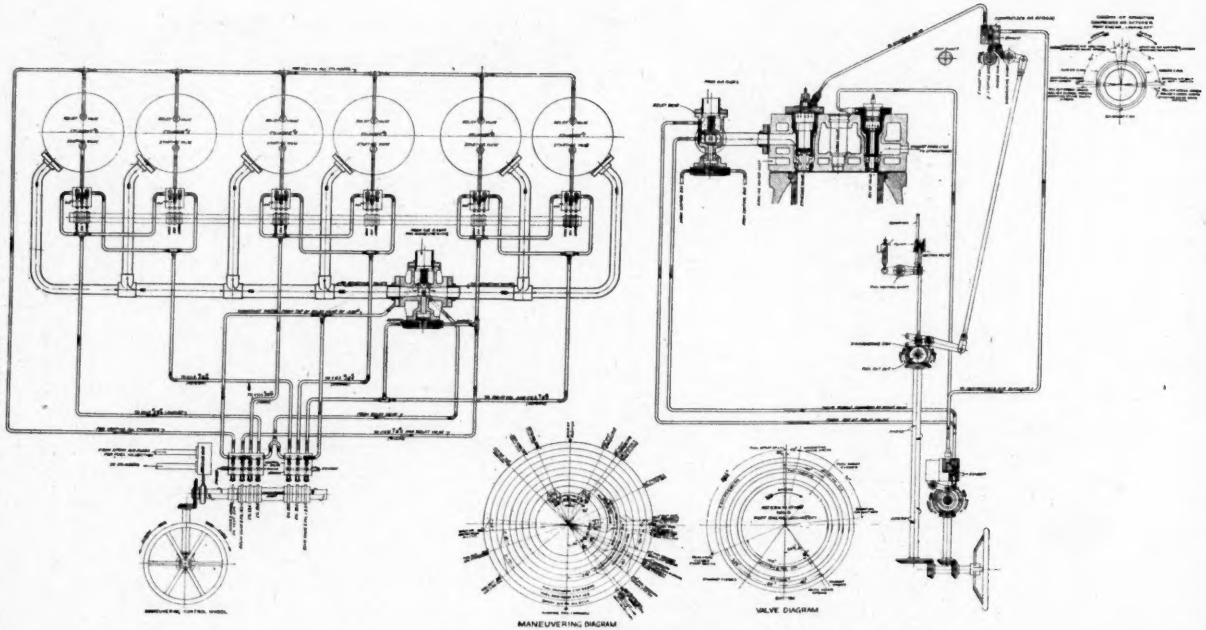
Each fuel pump has a mechanically-operated suction valve and two discharge valves in series. The pump delivers through a fuel-check valve held shut by spring and spray air pressure to the fuel-spray valve body in the center of the cylinder head. The same crosshead that drives the fuel-pump plunger also operates the lever that opens the suction valve. The inboard end of this lever acts as the fulcrum, and can be lowered or raised at will from the operator's platform, thus varying the period of opening of the suction valve. The speed and power of the engine are controlled by varying the period of opening of the suction valve, and hence the quantity of fuel pumped to each cylinder.

Before proceeding further it is proper to describe the cycle of events in the working cylinders of the engine on starting and in operation. Compressed air at about 650 pounds per square inch from the air-starting bottles is led to those cylinders whose cranks are in the proper position for running in the desired direction. After the engine starts to turn, starting air is admitted to each cylinder from 10 degrees past top center to 85 degrees past top center until the engine has attained sufficient speed for fuel admission. This speed is usually obtained in a very few revolutions. The operation of starting is quickly and easily accomplished in less than ten seconds, as will be explained later, the entire control being centered in one operating or "maneuvering wheel."

With the engine running normally, fuel is admitted to the cylinder once every revolution from 2½ degrees before top center to 37½ degrees after top center. Prior to the fuel admission clean air from the scavenger receiver has been compressed to about 450 pounds per square inch, at which pressure its temperature is about 900 degrees F. The fuel-spray valve is opened by the cam shaft through a lever, admitting the fuel to the cylinder, when the heat of the compression is greatest. The fuel enters in the form of a finely



SECTIONAL ARRANGEMENT.



AIR STARTING DIAGRAM.

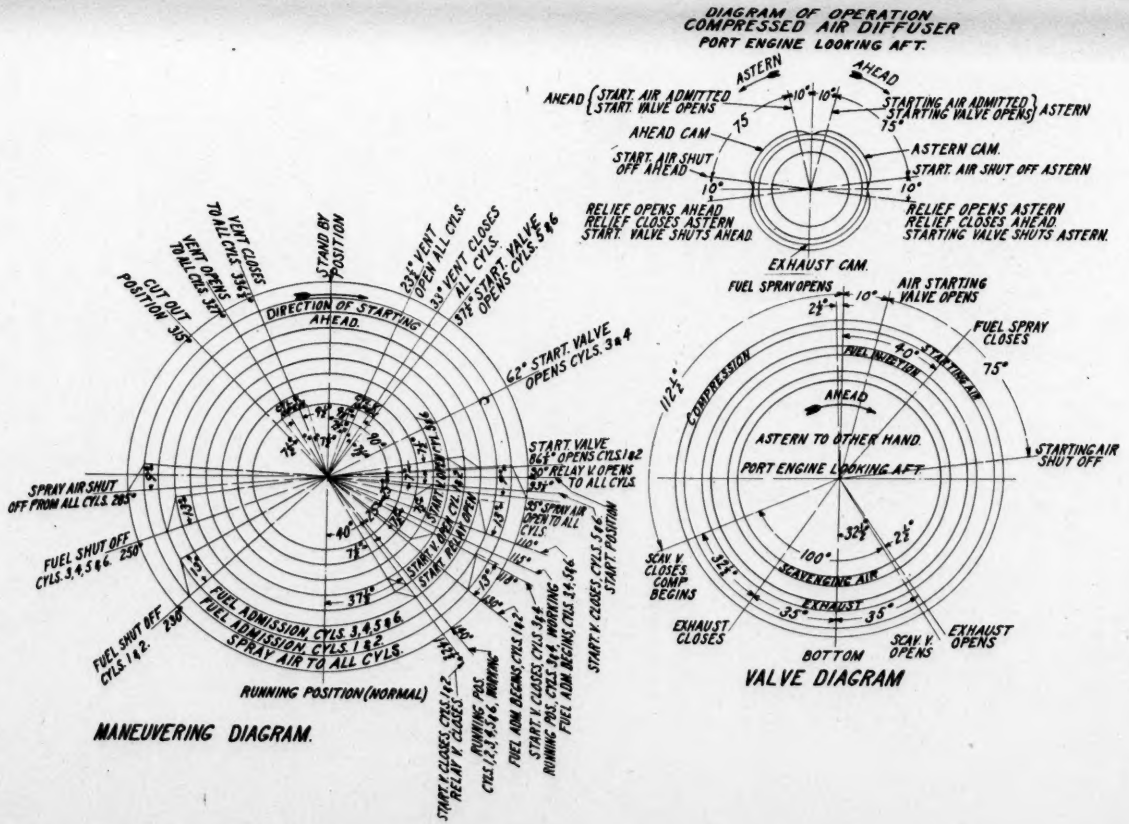


DIAGRAM OF OPERATION
COMPRESSED AIR DIFFUSER
PORT ENGINE LOOKING AFT.
ASTERN
AHEAD
AHEAD {START. AIR ADMITTED START. VALVE OPENS}
10°
10°
{STARTING AIR ADMITTED STARTING VALVE OPENS} ASTERN
75°
75°
AHEAD CAM.
ASTERN CAM.
START. AIR SHUT OFF AHEAD
START. AIR SHUT OFF ASTERN
10°
10°
RELIEF OPENS AHEAD
RELIEF CLOSES ASTERN
START. VALVE SHUTS AHEAD.
RELIEF OPENS ASTERN
RELIEF CLOSES AHEAD
STARTING VALVE SHUTS ASTERN.
EXHAUST CAM.
FUEL SPRAY OPENS
AIR STARTING VALVE OPENS
10°
2½°
FUEL SPRAY CLOSES
40°
STARTING AIR
75°
FUEL INJECTION
COMPRESSION
112½°
AHEAD
ASTERN TO OTHER HAND.
PORT ENGINE LOOKING AFT.
STARTING AIR SHUT OFF
SCAV. V. CLOSES COMP BEGINS
32½°
2½°
100°
SCAVENGING AIR
EXHAUST
32½°
35°
35°
EXHAUST CLOSES
EXHAUST OPENS
SCAV. V. OPENS
BOTTOM
VALVE DIAGRAM
STAND BY POSITION
VENT CLOSES TO ALL CYLS. 336°
VENT OPENS TO ALL CYLS. 324°
CUT OUT POSITION 315°
DIRECTION OF STARTING AHEAD.
23½° VENT OPEN ALL CYLS.
33° VENT CLOSES ALL CYLS.
57½° START. VALVE OPENS CYLS. 5 & 6
62° START. VALVE OPENS CYLS. 3 & 4
START VALVE 66½° OPENS CYLS. 1 & 2
50° RELAY V. OPENS TO ALL CYLS.
35° SPRAY AIR OPEN TO ALL CYLS.
START. V. OPEN CYL. 1 & 2
START. REACH OPEN
SPRAY AIR SHUT OFF FROM ALL CYLS. 285°
FUEL SHUT OFF CYLS. 3, 4, 5 & 6. 250°
FUEL SHUT OFF CYLS 1 & 2. 230°
FUEL ADMISSION CYLS. 3, 4, 5 & 6.
FUEL ADMISSION CYLS. 1 & 2
SPRAY AIR TO ALL CYLS.
START. POSITION
START. V. CLOSES CYLS. 5 & 6
FUEL ADM. BEGINS CYLS. 5 & 6
RUNNING POS. CYLS. 3 & 4 WORKING
START. V. CLOSES CYLS. 3 & 4
FUEL ADM. BEGINS CYLS. 1 & 2
RUNNING POS. CYLS. 1, 2, 5 & 6. WORKING
START. V. CLOSES CYLS. 1 & 2
RELAY V. CLOSES
RUNNING POSITION (NORMAL)
MANEUVERING DIAGRAM.

divided spray, or mist, and begins to burn at once, the spray continuing until the fuel valve closes at 37½ degrees past top center. Expansion now occurs until 35 degrees before bottom center, when the piston uncovers a belt of exhaust ports in the cylinder. Two and one-half degrees after the exhaust starts to open, two scavenger valves in the cylinder head are opened by the cam shaft, admitting clean air at about seven or eight pounds pressure per square inch from the scavenger receiver. This clean air blows the burned gases out through the exhaust ports and fills the cylinder with clean air for another cycle of events. The exhaust ports are again covered by the piston at 35 degrees past bottom center and compression occurs. The scavenger valves remain open until 32½ degrees after exhaust ports are closed by the piston. Compression now occurs until 2½ degrees before top center, when the fuel valve opens.

The operation of reversal consists merely in cutting off all fuel from the engine, which causes it to stop very quickly, and then to start in reverse direction by compressed air as described above.

For the purposes of this description the engine will be divided as follows:

The main engine, consisting of six working cylinders;
The high-pressure air system;
The scavenging air system;
The salt-water cooling system;
The fresh-water cooling system;
The low-pressure lubrication system;
The high-pressure lubrication system;
The fuel system;
The maneuvering and control system.

THE MAIN ENGINE.

The main-engine bed plate has open crank pits and is of the type ordinarily used on marine reciprocating engines. There are three cast-iron sections bolted together, each section being

cored and ribbed for lightness and extreme stiffness. The crank pits being open, provision will be made on the ship for an oil basin underneath the engine. Each section has pads on its top side for bolting four uprights or columns, and on its outboard after side for securing the attached pumps under the scavenger cylinders.

There are three main bearings in each section of bed plate, the middle bearing being about 3½ inches longer than the end bearings, which are about 16½ inches long. Each main bearing consists of a flat-bottomed cast-iron seat, supported in the bedplate saddle; a lower main bearing brass cored for water circulation and capable of being rolled out of the saddle without removal of crank shaft; a flat topped upper-bearing brass which is cored out for stiffness and lightness; and a forged-steel binding cap. The bearing brasses are lined with a white metal of the following composition:

Tin	80 per cent.,
Antimony	15 per cent.,
Copper	5 per cent.,

this being somewhat harder than Navy-standard bearing metal. Four studs are provided for securing the forged-steel bearing cap. The flat-bottomed cast-iron bearing seat is capable of vertical adjustment to take up wear in main bearings, and the two bearing brasses are lipped at each end to prevent fore-and-aft movement.

The thrust bedplate is of cast iron, cored and ribbed for stiffness and bolted to the after section of the main-engine bedplate. It has a well at its forward end for the flywheel and engine-turning gear, the latter being driven by an electric motor just outboard the flywheel. The thrust block is of cast iron and of the ordinary marine type, and is secured to the table on the after end of the thrust bedplate. The block is bolted to the bedplate by clearance bolts and is capable of fore-and-aft and vertical adjustment. The block is prevented from fore-and-aft movement by two tapered keys fitted at

each end between the block and the thrust-bearing base. The thrust block carries seven cast-iron thrust collars of the horse-shoe type, cored for water circulation and lined on the bearing faces with the special white metal mentioned above. The collars are capable of individual fore-and-aft and vertical adjustment on the thrust block, and are secured from fore-and-aft movement by a heavy, threaded rod on each side of the block. The thrust shaft, which is 14.567 inches in diameter, is supported in bearings at each end of the thrust block.

The main crank shaft, in three interchangeable sections, is 16.535 inches in diameter and is made of special forgings of the following characteristics:

Tensile strength, 71,000 to 78,000 pounds per square inch;
Elongation, 18 per cent. to 20 per cent.
Bending-test bar to be bent around an angle of 180 degrees cold.

The sections are hollow bored and drilled for the forced-lubrication system, the oil entering the main bearing journals from the lower main bearings. The sections are so bolted together that the sequence of the cranks turning outboard is 6, 1, 4, 5, 2 and 3. Counter weights are bolted and keyed to each crank shaft opposite the crank pin, and a flywheel, used also as part of the turning gear, is secured to the after end of the after section. The flywheel is made in halves, keyed and bolted together, in order to facilitate handling in the ship if it ever becomes necessary to remove it.

The main driving gears for the vertical shaft consist in a cast-steel helical gear made in halves and bolted around the coupling between the middle and after sections of the crank shaft, and a phosphor-bronze gear keyed to the bottom of the vertical shaft. The space between the middle and after sections of bedplate is made into an oiltight well for the gears to run in, the lower end of the vertical shaft being supported on the after side of the well. The lower bearing or support for the vertical shaft is itself secured in an eccentric bearing

set into the after bedplate, the object being to provide a means of taking up the wear between the steel and bronze gears.

The vertical shaft is on the inboard side of the engine and is in three sections, having keyed on its upper end a phosphor-bronze helical gear which drives the cast-steel gear on the cam shaft. The middle section of the vertical shaft has just below the upper coupling thrust collars which support the entire weight of the vertical shaft in a ball thrust bearing, which is in turn carried in a swivel bearing bracket bolted to the engine framing. The vertical shaft is again supported in its upper section by a swivel bearing bracket bolted to the steel casting forming a casing around the cam shaft and vertical shaft gears. The swivel bearing brackets are fitted with shims to provide for easy alignment.

The engine columns are made of cast iron, cored and ribbed for lightness and stiffness and are of the box pattern. The inboard columns carry the main crosshead guides, while the outboard columns carry the compressors for fuel injection and cylinder scavenging and the small crosshead guides for the auxiliaries. Outboard column No. 1 carries on its after side the fuel-supply pump driven off the crosshead of No. 1 cylinder and supplying fuel from the ship's bunker tanks to the fuel-supply tanks in the engine room. The outboard columns also carry on their forward and after sides the bearings for the beams. Inboard columns No. 3 and No. 4 carry on the lower platform all of the operating gear for handling the engine. Between columns 2 and 3, and 4 and 5, both inboard and outboard, are fitted cast-iron distance pieces to tie the columns in a fore-and-aft direction, forming a very rigid engine structure.

The cylinder bases are heavy iron castings, cored and ribbed, that tie the columns together athwartships and support the main working cylinders, these latter being secured by nuts and studs to the cylinder bases. The cylinder base has a hole in its center to allow the cylinder liner to extend below it.

The main crosshead slipper is of the usual marine type and is of cast steel, lined with hard white bearing metal.

The connecting rod and main crosshead, as well as the crank-pin and crosshead bearing brasses, are of the usual marine type, except that the bearings are lined with the special bearing composition and that the crossheads are bored out for the fresh-water passages of the piston-cooling system. The main crossheads have extensions fore-and-aft for driving the auxiliaries back of the engines by means of links and beams.

The piston rod is of forged steel and is hollow bored for the passage of the fresh water to and from the working piston. A nickel-steel pipe in the center of the piston rod conveys the water to the working piston while its return is effected by the concentric passage around the nickel-steel pipe. The upper end of the piston rod terminates in a heavy flange to which the working piston is securely fastened by means of nuts and studs. The lower end of the rod terminates in a square bolting flange for securing to the crosshead. A large round flange at about its middle serves to secure the lower part of the piston, which acts as a guide for the working piston.

The piston is divided into two parts, the working piston which consists of a special iron casting cored for water circulation and ribbed for strength; and a lower iron casting which is bolted to the piston rod. The two parts are not bolted to each other, although both are secured to the rod. The working piston is dished on top, and is machined with greater clearance at its top than at its bottom. It carries six cast-iron snap piston rings in grooves, which vary in width from the top on to the bottom one. The upper rings are given more clearance than the lower ones, on account of the greater heat. The fresh water coming up from the rod enters the central compartment of the piston, passes out towards the sides through cored passages at the top of the piston, and finally reaches the concentric space in the piston rod, via four pipes set at angles of about 45 degrees, returning from the

highest points of the water space, thus insuring a flow of water along the hottest parts of the piston. This method of conveying the cooling medium to and from the piston has the advantage of simplicity, but it has also the disadvantage of heating the water entering the piston by that just leaving the piston, and vice versa; the water just leaving the piston will not give a true indication of the piston temperatures as it has been somewhat cooled by the entering water. Eight large plugged holes in the bottom of the working piston afford easy access to all parts of the water space for removal of scale.

The lower part of the main piston, as has been explained before, serves only as a guide for the working piston. It is a cylindrical casting bolted at its lower end to a round flange on the piston rod. Its upper end is loose and free to expand, and fits over the lower end of the working piston. The guide piston is machined straight, with no taper to provide for expansion. Two cast-iron snap piston rings at the bottom of the guide piston prevent the escape of gas into the engine room. At the bottom of the stroke the top of the working piston just clears the ring of exhaust ports in the cylinder liner. At the top of the stroke the lower end of the guide piston just covers the exhaust ports.

The main cylinder is made up in two parts, a cast-iron jacket carrying the exhaust belt at its middle and a plain cylindrical liner of special cast iron.

The cylinder jacket carries twelve studs in its top flange for securing the cylinder head and a bolting flange near its bottom for securing to the cylinder base. At its middle it carries an exhaust passage which connects the cylinder exhaust ports with the exhaust piping. The space between the cylinder jacket and the liner forms the water jacket through which salt cooling water passes. The liner, which is thicker in section at its upper end, is pressed into the jacket until a shoulder on the liner rests on a shoulder in the upper part of the jacket. The top of the liner is securely held in place by the cylinder head while the lower end is free to expand

through the stuffing box in the bottom of the jacket, which prevents salt-water leakage. The bottom of the liner carries a double ring to catch salt-water leakage outside the liner or dirty oil from the inside of the liner. The oil and water thus caught is led to the bilge, from whence it can be pumped overboard. The liner has rectangular exhaust ports machined in a belt near its middle for the escape of the exhaust gases. The jacket is provided with eight handholes set 45 degrees apart, four in the upper part and four in the lower part, for the cleaning of scale from the liner. The jacket and cylinder are pierced at eight points 45 degrees apart for lubrication, four openings being in the upper part and four in the lower part of the cylinder. The press fit between liner and jacket is entirely at the exhaust belt, where every effort must be made to prevent the leakage of water from the cooling space into the exhaust ports. The surface of the liner passing through the tight fits at the exhaust belt has several shallow grooves whose object is to collect any slight water leakage, and "rust up," forming a watertight joint between liner and jacket. The surface of the jacket at the tight fits has annular grooves, one above and one below the exhaust belt. These grooves are about ¼ inch deep and ½ inch wide, and they are connected to pet cocks on the outside of the cylinder jacket. These pet cocks, kept open, act as leak indicators, should the water get by the other preventive measures. In case of leaks through the indicators, the proper procedure would be to gun the annular passages full of red lead, which would probably stop all leakage. A drain is provided at the lowest part of the water jacket. The cylinder liners are tested to 1,000 pounds per square inch above the exhaust ports, before being pressed into the jackets. A groove in the top of the liner forms a seat for the tongue on the bottom of the cylinder head. On the outboard side of the jacket is bolted an exhaust header of cast iron, cored for water circulation. Between the exhaust headers, sections of built-up exhaust piping are fitted, each section consisting of an inner and outer lap-welded steel pipe,

riveted to cast-steel flanges. The space between the inner and outer pipes forms the water jacket. The forward end of the exhaust pipe is closed by a water jacketed blank flange, while the after end is connected to the piping leading to the muffler. The exhaust headers and sections of piping are plentifully supplied with hand holes for cleaning the water spaces. On the inboard side of the cylinder jacket is bolted a bracket supporting a mechanical lubricator of the Detroit type. Two cast-iron brackets are bolted on the inboard side of each jacket near the top to support the cam-shaft bearings.

The cylinder heads are bolted to the cylinder by means of twelve long studs, the joint between the head and liner being made tight by a thin copper gasket, laid in the groove in the cylinder liner. The head has five openings in its top to receive the valve cages. The center opening is for the fuel-spray valve. The two largest openings are on either side of the spray valve in a fore-and-aft line and are for the scavenger valves. The inboard opening is for the cylinder-relief valve, while the outboard opening is for the air-starting valve. The scavenging-air receiver, consisting of a large copper pipe, runs the full length of the engine just outboard of the cylinders and near the heads. The receiver is supported by two pipes leading down to each scavenger pump, and by two pipes leading up to each cylinder head at the side, one for each scavenger valve in the head. The starting-air line runs the length of the engine and has a connection to each cylinder head at the side for the starting valve. The fuel-spray valve and the two scavenger valves are the only ones operated by the cam shaft. The starting valve is pneumatically controlled by the maneuvering wheel, but it may also be opened by hand. The relief valve is set to blow at 710 pounds per square inch, but it is also pneumatically operated by the maneuvering wheel.

Three types of cylinder heads are being tried on the first engine to be constructed. One type is shown on the general arrangement drawings, and is a special iron casting in one piece. The head is divided into two compartments for cool-

ing. The water from the cylinder jacket is by-passed around the cylinder-head joint into the lower compartment of the head, through which it must all go before rising to the upper compartment. A cast-steel sleeve in the center of the head receives the spray-valve body. Another type of head being tried is of the same design as that just described, but it has a much thinner bottom and is made of manganese-brass. The third type of head consists of a cast-iron upper story with a forged-steel bottom, the forged-steel bottom being machined out for water circulation. All the heads are drilled to receive pressure indicators and all water spaces are easily accessible for cleaning.

The relief valve consists of a forged-steel poppet valve, opening upwards, either by excess pressure in the cylinder or by compressed air acting on the bottom of the plunger attached to the valve spindle. The valve is carried in a separate valve body of cast iron, and is held shut by a spring. The upper part of the valve body contains a cylinder in which works the plunger attached to the valve stem.

The starting valve consists of a forged-steel poppet valve carried in a separate valve body of cast iron, the upper part of the body acting as a cylinder in which operates a plunger attached to the starting-valve stem. The valve opens downward, due to the admission of compressed air into the valve body on top of the plunger on the stem. The starting valve can also be opened by hand. The valve is ordinarily held shut by a spring.

The scavenger valves consist of forged-steel poppet valves, opening downward, due to the operation of the cam shaft and the scavenger levers. The valves are ordinarily held shut by springs acting on the bottom of a plunger or guide on the valve stem. The scavenger valves are housed in cast-iron valve bodies.

The fuel-spray valve, located in the center of the head, consists of a cast-iron body housing a long forged-steel needle valve opening upwards. This valve is opened by the cam

shaft and it is ordinarily held shut by a heavy spring. The compressed air for fuel injection is connected to the valve body at the top and maintains a constant pressure in the body, there being a safety valve in the air line at each cylinder. The fuel-oil line from the cylinder measuring pump is also connected to the top of the valve body, but fuel flows into the valve body only on the down stroke of the pump plunger, and then only when the engine is in the "operating" condition. As the fuel must be delivered to the valve body against the compressed air for injection, a check valve is installed in the fuel line as close to the valve body as possible. As it is extremely important that this fuel line be free of air at all times, a by-pass valve is fitted in the fuel line close to the fuel check but between the pump and the check valve. This by-pass permits the fuel to flow into a sight cup at the operator's station, and an inspection will indicate at once whether or not the fuel check leaks or whether the line is clear of air. The continuous presence of bubbles in the fuel from the by-pass indicates leaky fuel check, which must be ground in before the engine will operate properly. A small hand pump in the fuel line to the engine can be used preparatory to starting for priming the fuel system so that the engine will start on fuel with as few revolutions as possible. Whenever the fuel-spray-valve is raised, the fuel and air in the valve body are forced to pass through a series of perforated discs and by a cone-shaped atomizing device resembling a conical reamer. The mixture of air and fuel, after passing by the valve seat of the spray valve, enters the cylinder through an orifice in a spray button on the bottom of the valve body. The opening into the cylinder is cone-shaped with a view to injecting the mechanically-mixed air and fuel to all parts of the combustion space.

The inboard side of the cylinder head supports two stanchions which carry the shaft for the three valve levers for each cylinder, the middle lever operating the fuel-spray valve, while the other two operate the scavenger valves. Along the tops of the spray valves runs a shaft, which is turned by hand

from the operator's station. This shaft, by means of a worm and worm wheel in the top of each fuel-valve body, runs a stop up and down to regulate the stroke of the fuel needle. The fuel-valve lever in raising the valve acts on the heavy spring which holds the valve shut. When the stop is run down the fuel lever, after raising the valve the desired amount, merely compresses the spring without further movement of the valve.

The cam shaft is on the inboard side of the engine and is in four sections. The first section (from forward) carries the cams for cylinders No. 1 and No. 2. The second section carries the governor, the cams for cylinder No. 3, and an eccentric for driving fuel pumps for cylinders No. 1 and No. 2. The third section carries the cams for cylinder No. 4 and the gear which transmits the motion of the vertical shaft to the cam shaft. This section also carries a thrust collar to prevent lateral movement of the shaft. Section four carries the cams for cylinders No. 5 and No. 6. The cam shaft is supported in bearings contained in cast-iron casings mounted on the right-angled brackets on the inboard sides of the cylinders, two brackets being under each casing. The gear casing around the cam-shaft gear is somewhat larger than the others and is supported on a vertical box column bolted to the engine framing. There are three cams for each cylinder, two for scavenger valves and one for the fuel-spray valve. The cam shaft turns at the same speed as the crank shaft, but in the opposite direction. As the engine is reversible, provision must be made for utilizing the same set of cams for both ahead and astern operation. The cam-shaft gear is not keyed to the cam shaft, but it carries the male part of a two-jaw clutch. The female part of this clutch is keyed to the cam shaft and the female jaws are 35 degrees wider than the male jaws. Assume that the engine is running ahead and it is desired to reverse. The engine is first brought to a stop by cutting off the fuel and spray air. Compressed air from the starting flasks is then admitted to those cylinders whose cranks are in the right po-

sition for starting astern. The crank shaft moves through 35 degrees before the cam shaft starts to turn, when the cams will be in the same positions for running astern as they were for running ahead. The advantage of this method lies in its simplicity and the necessity for only one cam for each valve for both ahead and astern. Some engines have two sets of cams, one set for ahead running and one set for astern running. In this latter case the levers must be raised off the cams while the cam shaft is moved along its length to engage the second set of cams. The scavenger cams are solid steel forgings, case hardened, while the fuel cam is provided with an adjustable hardened steel toe, which permits of slight variations in the timing of the spray valves. The cams operate the valve levers through hard composition rollers in the inboard ends of the levers.

Bolted to the side of the cam-shaft gear is a smaller gear, which drives the starting shaft, just inboard of the cam shaft. As the cam-shaft gear turns at the same speed as the crank shaft, but opposite in direction, so the starting shaft turns in the same direction, as the crankshaft, and *always* with it. This shaft is mounted in bearings held in the cam-shaft casings and carries three cams directly underneath a distribution block for each cylinder. One cam operates a poppet valve in the distribution block for ahead motion of engine; one cam operates for astern motion, and the middle cam serves to vent the air piping which opens the starting valve. The distribution blocks are composition blocks forming part of the air-starting system and serve to open the proper starting valve on the cylinders at the proper time. As the engine starts to turn, the cams on the starting shaft, through the distribution blocks, hold open each cylinder-air starting valve from 10 degrees past top center to 85 degrees past top center, until the distribution blocks are rendered inoperative by means of the maneuvering control wheel.

Just inboard from the starting shaft is another shaft, called the "disengaging" shaft. This shaft, in two sections, is

operated entirely by the maneuvering wheel and serves to put in operation the distribution blocks for starting in either direction Cams on the disengaging shaft operate on bell-crank levers under the distribution blocks. The outboard ends of the bell cranks contain rollers which transmit the motion of the *starting-shaft* cams to the poppet valves in the distribution blocks. A flat spring under each block serves to hold the inboard ends of the bell cranks against the cams on the disengaging shaft. When the engine is running normally the cams on the disengaging shaft hold the rollers clear of the starting-shaft cams. The cams on the starting shaft, one for ahead, one for astern and one for venting the system for each cylinder, are so set that at least one crank will always be in the proper position to start the engine in the desired direction the instant starting air flows to the starting valves. The operation of the maneuvering wheel permits air to flow only to that end of the distribution block which will start the engine in the desired direction, the other half of the block being held open to the atmosphere.

THE HIGH-PRESSURE AIR SYSTEM.

The high-pressure air system of one engine consists of the three attached air compressors, the spray flask of about five cubic feet capacity, the six starting-air flasks with capacity of about 180 cubic feet, air separators, piping, relief valves, etc. One auxiliary air compressor, independently driven by steam, with capacity equal to that of one of the attached air compressors, is also provided for charging spray and starting flasks when all air is gone. The auxiliary compressor is of the three-stage, tandem-piston, slow-speed, horizontal type as manufactured by the Ingersoll-Rand Company. The suctions of the low-stage cylinders of all compressors will be taken from the coolest parts of the engine room or from outside the engine room if necessary on account of the reduced efficiency caused by sucking in high-temperature air. As the low-pres-

sure stage of the attached compressors is double acting, both suction pipes lead to a common valve which controls, via the operator's platform, the admission of air to the compressors. These suction valves can also be regulated at the compressors, independently of the gear at the operator's platform, so that any one of the three attached compressors can be made the reserve compressor. The valves of the low and intermediate cylinders are carried in detachable valve cages, making them very accessible for overhaul or examination. The valves of the high-stage cylinder are in the cylinder head and are easily accessible. The compressed air is cooled by its coil in the intercooler after each stage and is finally delivered to the collecting pipe back of the engine through a non-return valve. The collecting pipe, having received the air from all the compressors, delivers to a separator, where oil and water are partially removed. From the separator the air is led to an air manifold, into which all attached compressors of both engines and the auxiliary compressor supply the air. From the manifold the air may be led entirely to the spray flask or to the starting bottles or partially to each, the spray air being kept at the desired pressure while the balance of the compressed air is stored in the starting bottles. From the manifold the air for fuel injection is led to the spray flask. The spray-flask head is provided with a spring-loaded overflow valve, which permits any excess over the desired spray pressure to flow to the starting bottles via the manifold. The spray flask is also provided with a drain for freeing the air of oil and water. From the spray flask the air for fuel injection is led to an automatic cut-out valve at the control station, the object of which is to have the air on the injection system only when the engine is in the running condition. During reversal the spray air is cut off the engine until the crank shaft has started to turn in the desired direction. From the automatic cut-out the air is led to a manifold pipe at the top of the engine, from which a short lead supplies each fuel-spray valve body. A relief valve set at 1,200 pounds per square inch and a check

valve are placed in the line to each fuel-valve body, as close to the valve body as possible.

THE SCAVENGING AIR SYSTEM.

The scavenging or low-pressure air system consists of the three attached double-acting scavenger compressors, with their piping, relief valves, etc. The air, drawn from the engine room or from outside, passes through the suction and discharge valves of the scavenger compressors into the coolers, two in each scavenger cylinder casting. Each cooler consists of a nest of straight tubes with cooling water inside the tubes and air outside. From the coolers the air is led to the scavenger receiver, a large copper pipe running the length of the engine. The receiver is made up in three sections, with an expansion joint between the sections. At each end of the receiver is a large relief valve set at 15 pounds per square inch. Each expansion joint is fitted with a drain for the removal of water or oil in the air. From the receiver two connections to each cylinder head convey the scavenging air to the scavenging valves.

THE SALT-WATER COOLING SYSTEM.

The salt-water cooling system consists of two attached plunger pumps, under the middle scavenger pump, and an independently driven steam-plunger pump, together with the necessary connections and piping on the engine. The independently-driven steam pump will also be connected up to run by compressed air until steam can be gotten up on a boiler.

Both attached pumps have a common suction, each pump being of sufficient capacity to supply the salt-water system at normal power. The salt water is discharged into a large main back of the engine, underneath the floor plates. From the main a branch leads upwards to the bottom of each intercooler for the high-pressure air compressors and to the bottom of each cooler in the scavenger pump castings, making a total of nine outlets from the main back of the engine. The

main continues around the forward end of the engine, where a branch leads upward and aft on the outboard sides of the main bearing caps. This branch supplies water to the jackets of the lower main bearings and to the thrust collars on the thrust block.

The supply main continues around to the inboard side of the engine under the floor plates, a branch being led to the bottom of each ahead crosshead guide. A collecting main runs around the engine at the heighth of the cylinder bases. On the inboard side of the engine it receives the return cooling water from the crosshead guides. At the after inboard side it receives the return cooling water from the main bearings and thrust block. On the outboard side of the engine it receives the cooling water from the scavenger coolers. Back of the engine all of the water in the collecting main enters the bottom of the main cylinder jackets, two branches leading to each jacket. The cooling water leaving the high-pressure intercoolers of each compressor is led to the lower end of the jacket of the middle-stage air-compressor cylinder. From here it is forced upward into the jacket of the low-stage cylinder through two ferrules set partly into each cylinder at the joint. From the low-stage jacket the water enters the high-stage jacket through two by-passes around the cylinder joint. From the high-stage jacket the water is forced into the high-stage cylinder head, through two by-passes around the joint between head and cylinder. From the head of each high-stage cylinder the water is led into the exhaust-pipe jacket. This return water was originally forced into the above-mentioned collecting main from whence it entered the main cylinder jacket, but the heating of the compressors due to poor circulation of the water forced the change described above. From the main cylinder jackets the water enters the cylinder head through a by-pass around the joint between the cylinder and the head. After circulating through the lower and upper compartments of the head, the water enters the exhaust-pipe water jacket. From the exhaust-pipe jacket the

water is finally discharged into an overboard-discharge main. A small pipe leading from the cooling outlet of each cylinder head to a pet cock discharging into a funnel at the operator's station, indicates whether or not the cooling system is functioning properly.

THE FRESH-WATER COOLING SYSTEM.

The fresh-water cooling system is supplied by the forward attached pump on the engine and can also be supplied by the independently-driven steam pump provided for emergency fresh-water circulation of both main engines. The fresh water is drawn from a double-bottom compartment where it is stored, and discharged into a large main back of the engine and below the floor plates. From the main a connection leads to the after beam bearing on each outboard column. At this point the water passes, via swivel joint, through the bearing to a pipe secured to the beam. This pipe conveys the water to the crosshead end of the beam, where it again passes to the after side of the beam, via swivel joint, through the lower bearing of the link. Another pipe with swivel joint conveys the water into the after end of the main crosshead, from which point the water reaches the working piston by means of a hole through the crosshead connecting to a nickel-steel pipe running up the center of the piston rod. The water is delivered into the center compartment of the working piston, from which it passes to the outer compartment through cored passages at the top. Four collecting pipes, reaching the highest parts of the outer cooling space, return the fresh water to the center of the piston rod into a space around the upward conveying pipe. From the piston rod the hot water reaches a discharge main back of the engine, via links and beams and the forward end of the crosshead, in a manner similar to that of entering. The discharge main returns this hot water to the double-bottom compartment from whence it came. The water will be cooled by the ship's bottom and by introducing cooling coils, if found necessary. A small copper pipe lead-

ing from the discharge side of each piston, is led to the inboard side of column No. 2, where it delivers through a small pet cock into a funnel. The operator on the handling platform is thus able to see at a glance whether or not the system is functioning properly. He can also *feel* the temperature of the cooling water. A thermometer is installed in the supply main near the pump discharge and additional thermometers are installed in each outlet from the pistons.

The emergency independently-driven steam pump in the ship for fresh-water cooling will also be connected up to the compressed-air system, so that the pump can be operated by air until steam can be raised on a boiler.

THE LOW-PRESSURE LUBRICATION SYSTEM.

The low-pressure lubrication system is supplied by the after plunger pump under the forward scavenger cylinder. The pump sucks from a well in the engine crankpit and discharges through a strainer, and a Schutte & Koerting film cooler to the main, which runs the length of the engine just inboard of the main bearing caps and underneath the floor plates. From the main a supply pipe leads to the bottom of each main bearing brass. The main also delivers to a manifold which supplies the thrust collars and bearings. The oil flows into a drilled passage in the lower main-bearing brass casting, from which it enters a fore-and-aft groove in the bearing surface through two supply holes. Each main journal has four lubrication grooves, 90 degrees apart, machined into its surface. Each groove connects to the center of the shaft, through a drilled hole. Two of the grooves on the journal collect the oil from the forward end of the groove in the lower bearing, while the other two collect the oil from the after end of the supply groove in the main bearing. From the center of the main journal the oil passes to the crank-pin bearings through passages in the webs, and drilled holes in the crank-pin journals. From the crank-pin bearings the oil passes through a hole in the top brass to a pipe leading to the lower bearing of

each crosshead brass. From the crosshead the oil drops back to the crank pit, or is caught at the ends of the bearings and returned to the crank pit by pipes. The oil from the thrust block is also drained back into the crank pit. The emergency-lubrication pump on the ship will be independently driven by air as well as by steam.

THE HIGH-PRESSURE LUBRICATION SYSTEM.

The high-pressure lubrication system is separate for each cylinder and is supplied by a 32-feed Detroit mechanical oiler mounted on the inboard side of each cylinder and driven by bevel gears from the starting shaft. Each oiler receives its supply from a connection to the low-pressure lubrication system. Each oiler has eight connections to each main cylinder, six connections to the main crosshead guide, three connections to the auxiliary crosshead guide, four connections to the scavenger cylinders on even numbered cylinders, and various connections to link and beam bearings, etc. The high-pressure compressors have special gravity-feed lubricators attached to each cylinder.

The cam-shaft bearings are lubricated by gravity-feed cups.

THE FUEL SYSTEM.

The fuel system is supplied by two gravity tanks for each engine, these tanks being kept full by the attached fuel pump driven off the after beam of No. 1 cylinder. This pump draws the fuel from the ship's bunkers and delivers through a strainer to the engine-room gravity tanks. From the gravity tanks the fuel flows through a filter to the suction side of each cylinder fuel pump. The speed and power control of the engine by means of the suction valves of these pumps has already been explained. The cylinder fuel pumps discharge to their spray-valve bodies through check valves close to the fuel-spray valves. A by-pass valve in the fuel line at the highest point just before it reaches the fuel check permits the

fuel line to rid itself of all contained air, and also serves to detect leaky fuel checks, as explained before. The fuel-supply tanks contain heating coils in case it becomes necessary to heat the fuel. An oil heater of the straight-tube type is also installed in the line between supply tanks and cylinder-fuel pumps, the circulating water from the cylinder heads being used to heat the oil. A small hand pump for priming the fuel system is also introduced in the supply line between the gravity tanks and the cylinder-fuel pumps. Before starting the engine, it is necessary to open all by-passes in the fuel lines to cylinder heads and, by means of the hand-priming pump, fill the fuel line so that one or two revolutions will force fuel into the spray valves.

MANEUVERING AND CONTROL SYSTEM.

On the operator's platform, on inboard columns 3 and 4, are located the devices for the power and speed control and the maneuvering of the engine. On the after side of column No. 4 is the maneuvering-control wheel, which controls the starting, stopping and reversal of the engine by means of compressed air.

This wheel also cuts out the fuel and spray air from the cylinders during the maneuver and until the engine is turning over in the desired direction. Above the maneuvering control is a dial on which a pointer indicates the running positions of the engine.

On the forward side of No. 4 column is the fuel-control wheel, which governs the quantity of fuel pumped into each cylinder. A pointer and dial above the fuel control indicate arbitrarily, in eight equal steps, the quantity of fuel pumped, from the minimum to the maximum.

Coming out from the shaft of the fuel-control wheel is the needle-stroke control, which varies the stroke of the fuel spray needle from the maximum to the minimum. On the after side of column No. 3 is a hand cut-out, by means of which the

engine can be instantly stopped. It operates to raise the suction valves of the fuel pumps, which renders them inoperative.

Below the hand cut-out is the control for the high-pressure air supply, which regulates the openings of the suctions of the low-stage cylinders of the air compressors. The quantity and pressure of the spray air is controlled by this control.

The future of the Diesel engine of moderate power and speed of revolution is assured on land and on slow-speed ocean-going freighters or tankers of moderate size. Ship owners are keenly alive to the advantages of the motor ship over the steam ship, this being evidenced by the many orders for motor ships being booked by the Burmeister and Wain Company of Copenhagen, Denmark. The war in Europe has left this company in the enviable position of being the only company with motor-ship experience whose shops are available for commercial work.

The revival of the shipping industry of the United States will undoubtedly bring orders for motor vessels to those shipyards in this country prepared to build the new types of engines.

PRACTICAL LUBRICATION.

By Lieutenant G. S. Bryan, U. S. Navy, Member.

The average engineer can hardly be expected to make an extensive study of lubrication and there is very little literature on the subject that is written in a form that is of practical use to him. Nearly all articles on the subject are either too theoretical and technical or else they are written in the interest of a certain brand of oil, and the facts correspondingly distorted or misrepresented to fit that particular oil. This article is intended to give briefly and in as practical a way as possible the general information that is needed by the engineer to assist him with lubrication problems.

WHAT IS A LUBRICANT?

We all know that two metal surfaces rubbing together give rise to considerable friction and to reduce this friction we use a lubricant between them. The theory is that the oil forms a thin film that entirely separates the two metal surfaces, and the friction generated under these conditions occurs in the body of the lubricant itself, and is due to the molecules of oil rolling or sliding on each other. We might consider, as an illustration, that the two bearing surfaces float on the film of oil.

The fundamental characteristics that a lubricant must have then are

1st. It must be capable of forming a thin film between the two surfaces.

2d. It must resist being squeezed out from between them.

3d. It must have low internal frictional qualities.

Any substance that satisfies these three requirements can

be classed as a lubricant. There are other qualities that a *good* lubricant should have, but the above three are the fundamental ones. About the only substances that possess these qualities to a sufficient degree to be of practical use are the so-called mineral, animal and vegetable oils.

Formerly the animal and vegetable oils were used almost exclusively for lubricating engines and machinery. Of late years, particularly since the advent of forced-feed lubricating systems, mineral oils have come into general use on account of their cheapness as well as other properties which make them more suited for general lubrication. In the naval service all oils used are pure mineral oils except that used for marine engines without forced lubrication, this oil containing a small percentage of blown rape-seed oil.

VISCOSITY INDICATES THE FRICTION.

The characteristic that indicates the internal frictional qualities of an oil is called the *viscosity*. It is measured by the time required for a certain amount of the oil to flow through a small standard orifice in an instrument called a viscosimeter. The viscosity can be judged roughly by noting the sluggishness with which an oil flows, and two samples can be compared in this respect by suddenly inverting sample bottles containing them and noting the comparative rapidity with which the entrapped air bubbles rise to the surface. In general the more viscous the oil the greater the friction generated and the greater the ability to withstand being squeezed out from the bearing.

The viscosity of an oil varies with the temperature, however, and the friction will also vary correspondingly. The higher the temperature the lower the viscosity, until at about 300 degrees F. nearly all oils have a viscosity only a little greater than that of water. The variation with the temperature is shown graphically by the curves in Figure 1, which are made for typical oils.

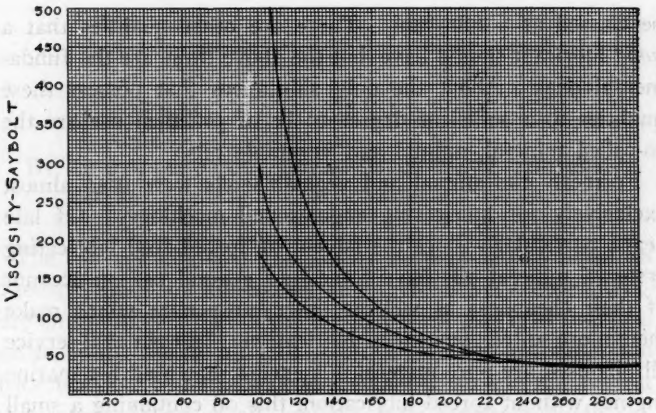


FIG. 1.—TEMPERATURE OF OIL,—DEGREES FAHRENHEIT.

THE EFFECT OF TEMPERATURE.

The first effect of a high temperature in a bearing, then, is to thin the oil and thus decrease the friction. The effect of cooling the bearing is to increase the friction. There is no virtue in keeping a bearing at a very low temperature by the use of a considerable amount of cooling water, and there is no reason why a bearing should not be allowed to run at a high temperature as long as it is in good condition and is getting plenty of good clean oil.

The principal reason why a hot bearing is so feared is that it is generally an indication that something is wrong with the bearing or that it is not getting sufficient oil. Possibly this is a good reason for not allowing bearings to run at very high temperatures, but there is no excuse for going too far the other way and keeping them too cool. There is a turbo-generator in the power plant of the Naval Engineering Experiment Station at Annapolis, the bearings of which run regularly at temperatures between 190 F. and 210 F., and this has never given any trouble, although the oils used in it have ranged from the heaviest forced-lubrication oil to the lightest ice-machine oil in use in the service. In one case an

average temperature of 205 F. was maintained for 30 days. Cooling a bearing below a fair working heat by flooding it with oil results in a waste of oil, and keeping it cool by circulating a large amount of cooling water around it results in a waste of power. However, in case a bearing heats up due to the presence of dirt or foreign matter, an increased supply of oil may be necessary to wash this dirt out of the bearing.

THE EFFECT OF THE LOAD.

Theoretically, the total friction in a perfectly lubricated bearing is independent of the load that may be put on the bearing. This may appear surprising, but it nevertheless is true, and can be easily demonstrated. In practice, while we do not find this rule to hold exactly true, we do find that it holds approximately so. The coefficient of friction drops very appreciably with an increase in the bearing pressure, the coefficient of friction being defined as the ratio between the

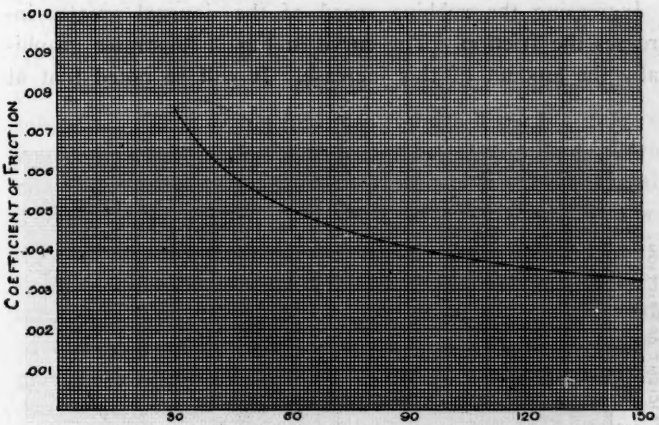


FIG. 2.—BEARING-CAP PRESSURE—POUNDS PER SQUARE INCH.

total friction and the total load. In Figure 2 a typical curve of the coefficient of friction is shown with points plotted for 30, 60, 90, 120 and 150 pounds per projected square-inch load

on the bearing, and this shows graphically the manner in which the above ratio varies with the load.

The principal danger of increased load on the bearing, then, is not that it directly increases the friction, but that it tends to squeeze out the oil from between the journal and bearing and thus destroy the lubrication. Generally speaking, the lighter (less viscous) the oil the easier it is to squeeze it out. It can be easily seen that the facility with which the oil is squeezed out naturally acts to increase the amount of oil used, since the oil forced out must be replaced by fresh oil. Forced lubrication is used mostly in the Navy now, and the loss would also be increased in this case by the more rapid circulation of oil through the system, since this would increase the leakage from imperfect joints, splashing of oil into the bilges by engine cranks, etc. This has caused many engineers to change to a heavier oil as a matter of oil economy.

THE EFFECT OF JOURNAL SPEED.

Increasing the rubbing speed of the journal greatly increases the friction. The curve in Fig. 3 illustrates graphically the amount of this increase. It will be noted that at

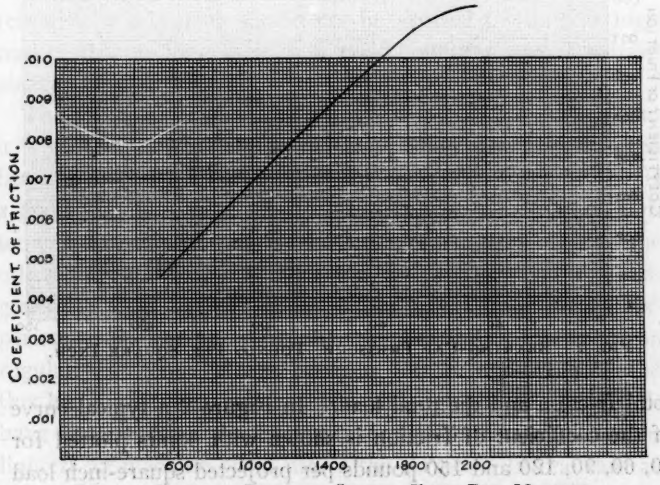


FIG. 3.—JOURNAL-RUBBING SPEED—FEET PER MINUTE.

rubbing speeds above 2,100 feet per minute the curve appears to flatten. This speed is exceptional, however, and at the speeds ordinarily encountered in the Service the friction increases greatly as the rubbing speed increases.

SELECTING A SUITABLE OIL.

In deciding whether to use a high (heavy), intermediate, or low (light) viscosity oil on a particular bearing we have three factors to consider—(1) temperature, (2) pressure, and (3) journal-rubbing speed; and we have just discussed what will be the effect of varying these factors.

The temperature that we should consider is the one we expect the bearing to have under practical working conditions and we should choose our viscosity accordingly. If we have two bearings that are identical, one working at 100 F., and the other at 150 F., we should have an oil of a certain viscosity at *100* F. for the first bearing and one of the same viscosity at *150* F. for the second.

With regard to pressure, the important thing is to get an oil that has body enough to prevent it from being squeezed out under the maximum load to which the bearing will be subjected. Then if you find that you are using too much oil you can generally reduce the amount by substituting a heavier oil. This will save oil but will increase the friction slightly. The oil finally selected must necessarily represent a compromise between the different factors.

In a high-speed bearing it is necessary to use a thin oil, on account of the great friction due to this high speed. Bearings of this type are generally designed so as not to have a very great bearing pressure, so that a light oil can be used without danger of it being squeezed out.

In turbine bearings the best conditions are obtained for the use of light oils. Here the speed is high and the bearing pressure is not only small, but is also steady and uniform in character. In a reciprocating engine the rubbing speed is much slower and the pressure on the bearings is intermittent

in character. The reciprocating action intensifies the bearing pressure and the squeezing out effect, but has the saving quality that while this pressure is on one side of the bearing the oil film is given an opportunity to form on the opposite side and be ready to meet this. On account of the above action that takes place in a large reciprocating-engine bearing we need a fairly heavy oil to give an increased "cushioning" effect and to prevent wasting the oil. The limit of viscosity in forced-lubrication systems is set by the inability of the oil pumps to handle a too heavy oil. With engines not fitted with forced-feed lubricating systems it is set by the ability to feed properly.

Turbines fitted with forced-feed systems give the nearest approach to perfect lubrication. In case the bearings heat up it may be due to grit getting in the bearing, and in this case the pressure on the system can be increased with the hope of forcing this out. The increase in pressure does not lessen the friction, though it does assist in carrying off the heat generated by friction and keeps the bearing slightly cooler.

PHYSICAL CONSTANTS.

As so much attention is often paid to physical constants, it might be well to make a few remarks here concerning them.

Viscosity.—We have already discussed this property and the effect of varying conditions on it; it only remains to describe how it is measured. The standard method is to express it as the number of seconds of time required for a given amount of oil to flow through a standard orifice. In the naval service the standard viscosimeter is the Saybolt, and the viscosity is generally expressed as so many seconds Saybolt at 100 F. Forced-lubrication and turbine oils, for instance, vary in viscosity from the lightest oils, which have a viscosity of about 160 seconds, to the heaviest, which have about 750 seconds at 100 F.

Flash Point.—This is the temperature at which vapors are given off in sufficient quantity to give a momentary flash when

a light is passed across the surface of the oil. For ordinary lubrication it is not of much value as long as it is high enough to be safe. Practically all lubricating oils have a flash point above 300 F. For the lubrication of steam cylinders a high-flash oil is necessary. For motor-cylinder lubrication a 300 F. flash point should be sufficiently high. This point was discussed at length in an article "Motor Cylinder Lubrication" in the February, 1915, number of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

Fire Point.—This is the temperature at which vapors are given off in such quantity that the oil will catch on fire and continue burning if a light is passed across its surface. Roughly it is generally about fifty degrees higher than the flash point. Except for this difference, its value as indicating anything about an oil is the same as that of the flash point.

Specific Gravity.—This gives no indication of the lubricating value of an oil. To the expert it will give some indication of the nature of the crude from which it was obtained. Many consumers, however, still buy their oil on specifications as to gravity with the mistaken idea that this is an index of its lubricating value.

Color.—The color of an oil gives no indication of its lubricating value.

Cold Point.—This is the temperature at which an oil becomes too viscous to flow. An oil should, of course, have a cold point much lower than the temperature of the room in which it works.

KINDS OF OIL FOR DIFFERENT PURPOSES.

The general qualifications of different kinds of oil that are in use in the naval service are as follows:

Marine Engine Oil.—This kind of oil is used on marine engines without forced lubrication, it being fed either by wick or sight feed. As the oil is only used once, it must be heavy enough to stay in the bearings and not be squeezed out easily

—otherwise excessive waste will take place. Also, since water often gets in the bearings or is splashed on the crosshead slides, some provision must be made to prevent the oil from being washed away from the bearing surfaces.

On account of the foregoing reasons, marine-engine oils are composed of mineral oils compounded with a certain percentage of animal or vegetable oil—generally about 15 to 20 per cent. of blown rape-seed oil. This is the only kind of oil used in the Navy that is not a pure mineral oil. Animal and vegetable oils maintain their viscosity with increase of temperature much better than mineral oils; also when stirred with water they form thick emulsions, or, to use the usual engineering term, they "saponify." The effect of this saponification is to thicken the oil and to make it more adhesive. On that account a film of mineral oil can be washed off of a crosshead slide much more easily than a vegetable or animal oil.

Tests are now being carried out at the Engineering Experiment Station to determine if it is not possible to obtain a straight mineral oil that will answer this purpose, as many of these oils emulsify very readily. This would reduce the price of this class of oils considerably.

Forced-Lubrication Oils.—This class includes *turbine oils,* the difference in the two being one of viscosity only. Conditions in forced-feed systems are quite different from those in sight or wick-feed systems. Since the oil is used over and over again, any that may be squeezed out from between the bearing surfaces does not represent a loss; also the oil can be supplied in any amount, thereby eliminating the possibility of an insufficient supply to the bearing. Therefore a lighter oil may be used than with the wick feed.

Water may get in the bearings, as mentioned before, but in this case any oil that is washed off will immediately be replaced by a fresh supply. There is no need of any "saponification" of the oil here. On the contrary, every effort should be made to exclude water, since the oil must be kept in con-

tinuous circulation, and the effect of a water leak soon becomes cumulative. One of the primary requisites of a forced-feed oil, then, is that it must not saponify and must separate readily from water. The viscosity of the forced-feed oils in the service range from about 150 seconds to 750 seconds Saybolt at 100 F.

Motor Cylinder Oils.—This class of oils was discussed at length in an article "Motor Cylinder Lubrication" in the February, 1915, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

Steam Cylinder Oils.—In steam cylinders the oils are subjected to the high temperatures of the steam, and if their flash point is not sufficiently high they will vaporize, and therefore cannot adhere to the cylinder walls. Also their viscosity must be great enough to withstand this high temperature without their getting too thin. A cylinder oil should, therefore, be selected with reference to the pressure and temperature that will be encountered.

When wet steam is used there is always some condensation on the cylinder walls, and this is liable to wash the oil off if a pure mineral oil is used. To prevent this it is usual to add a small percentage of tallow or lard oil to the cylinder oil to cause it to emulsify and stick to the surfaces similar to the action of a marine-engine oil. The objection to this is that the fatty oil (tallow) will get in the feed water through the condenser and liberate fatty acids in the boilers, thereby causing increased corrosion and scale. On this account only pure mineral oils are allowed to be used in steam cylinders in the naval service. In fact, practically no cylinder oil at all is used in the steam cylinders in the naval service, the wet steam itself furnishing what lubrication is necessary.

For superheated steam no condensation on the walls is available for lubrication and the need of oil is therefore much greater. There is no danger of the oil washing off, however, for the same reason, and a straight mineral oil should be satisfactory. The temperature of superheated steam is, of course,

higher than that of wet steam and a higher flash point is necessary.

Ice-Machine Oil.—Most of the ice machines in the naval service are of the dense-air type. Oil used in the expander valve chest and cylinder should have a very low cold point and should not have too great a viscosity at the low temperatures at which the expander cylinder works. In order to get these qualities it is necessary to use a light oil.

In the compressor cylinder and valve chest the temperature is higher. In some cases it is so much higher that ordinary ice-machine oil will not have sufficient viscosity. In a case of this sort a heavier oil should be used in the compressor cylinder and valve chest. Where the return air is cool enough, regular ice-machine oil can be used in both expander and compressor. It is a good plan to do this where possible, as there will, of course, be some leakage of the oil from the compressor side to the expander side and a heavy oil would be very likely to gum there.

For other working parts of the ice machine the rubbing speed is low and the pressures not very great. Marine-engine oil or forced-lubrication oil should be used with the best results.

Dynamo Oil.—As the load on the bearings of a dynamo is generally light and the rubbing speed fairly great, a light oil similar to a turbine oil should be used. If too large a quantity is used the amount can be reduced by using a slightly heavier oil—for instance, a forced-lubrication oil.

Air-Compressor Cylinder Oil.—Steam cylinder oils give the best results in air-compressor cylinders. Owing to the high temperature and compression, an oil of high viscosity is necessary, and this condition is best fulfilled by steam-cylinder oils.

WATER IN BEARINGS.

A good deal of misunderstanding exists on the subject of water in bearings. A mixture of oil and water will not give as good lubrication as the oil alone. This is shown graphi-

cally in the curve, Fig. 4, which was obtained by measuring the coefficient of friction of a marine-engine oil first alone and then with various percentages of water added to it, the mixture being stirred thoroughly to make it saponify. This curve shows that even small amounts of water in a bearing will greatly increase the friction.

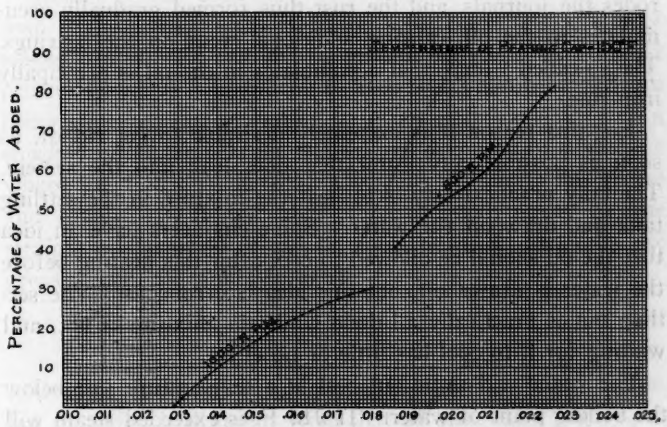


FIG. 4.—COEFFICIENT OF FRICTION.

This seems directly at variance with the old theory that a "lather" on a bearing is a sign of good lubrication. This theory was based on the fact that there was always a probability of water getting in the bearings and a corresponding fear that this water would wash the oil off the bearing surfaces and destroy the lubrication entirely. As froth on the bearing indicated that the oil and water would emulsify, this relieved the engineer's feelings to that extent.

The best plan is to use an oil that will emulsify and then keep water out of the bearings, remembering that if it does get in accidentally no great harm will be done. In cooling off a hot bearing the water should be applied through the water service and on the outside of the bearing, and care should be

taken not to mix it with the oil, as the cooling effect will be more than counteracted by the additional friction.

Forced-Feed Systems.—The foregoing remarks apply principally to sight and wick-feed lubrication. In forced-feed lubrication conditions are different, as previously explained, and emulsification is always objectionable. In addition to increasing the friction and giving less lubrication the water corrodes the journals, and the rust thus formed gradually accumulates in the oil, causing additional wear on the bearings. The so-called "dirt" that accumulates in the oil is principally iron rust.

For this reason it is extremely desirable to get rid of, as soon as possible, any water that does leak into the system. The only practical way is to heat the mixture in the settling tank and draw off the water. Some engineers have an idea that the oil should be allowed to cool after this heating before the water is drawn off, but this idea is erroneous. The settling out is practically all done while the mixture is hot, and waiting for it to cool has little or no effect.

The water will settle out best at a temperature just below the boiling point of water. If 212 F. is exceeded steam will form, and this will serve to agitate the mixture. A temperature of 200 F. should not have any bad effect on the oil. The oil should be cooled before being supplied to the bearings, however.

If marine-engine oil gets in the forced-lubrication system there is a tendency to form a thick, heavy emulsion which gives great difficulty in settling out. Some cases have been known where tallow, which had been put in bearings when a ship was laid up, was allowed to get in the forced-feed oil, and this resulted in the formation of a heavy emulsion which would not settle out.

Every effort should be made to get rid of the dirt in the oil and the strainers should be frequently cleaned. If water is kept out of the system most of the dirt will be automatically eliminated.

GREASES.

For the lubrication of piston rods or steam cylinders pure mineral steam cylinder oil should be used, and even then care should be exercised to keep it from getting in the condensers and boilers. Greases should never be used, as they will liberate fatty acids which eventually get in the boiler.

Greases consist essentially of a mineral oil thickened with soap. The lubrication is given by the oil and the soap simply makes it more solid and gives it greater body. They are used principally where it is difficult to keep an oil in a bearing from being squeezed out or where it is not practicable to give a steady feed to the bearing. They are also used to advantage where the frictional heat of a bearing will melt the grease and supply it automatically. It is a good policy to use straight oil wherever it can be done, however, and only use grease where oil cannot be used.

GRAPHITE.

Graphite is classed as a lubricant because it reduces friction, but its action is different from that of oils. If a highly-polished bearing surface is examined under a high-power microscope it will be seen that the so-called smooth surfaces are really quite rough, they really consisting of alternate ridges and depressions which are too small to be detected with the naked eye. If graphite is used on a bearing of this kind it will soon fill up all the depressions and give a bearing surface that is much smoother. This will, of course, reduce the friction. A steady supply, however, would soon clog the bearing.

Graphite and oil are often used together and will give good results where the graphite does not accumulate and clog the bearing. It often happens that the oil holes or pipes get stopped up in this way. The same objection applies to graphite grease.

U. S. S. *CUSHING.*

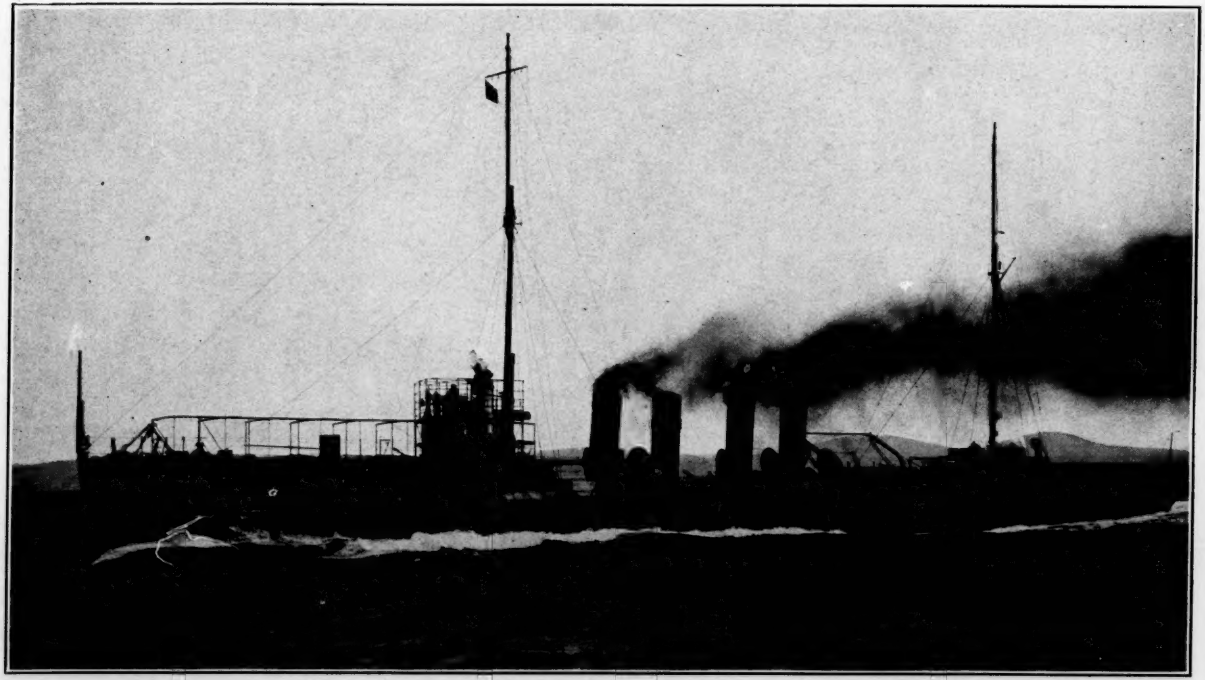
DESCRIPTION AND TRIALS.

By Lieutenant Ormond L. Cox, U. S. Navy, Member.

Torpedo-Boat Destroyer No. 55, the *Cushing*, is one of the six destroyers of the same class authorized by an Act of Congress approved August 22, 1912. The *Cushing* is a twin-screw vessel fitted with Curtis turbines, with geared cruising turbines and designed for a speed of 29 knots at about 1,050 tons trial displacement with the main engines developing about 16,000 shaft horsepower. The vessel was built under contract by the Fore River Shipbuilding Co. of Quincy, Mass. The contract was signed December 11, 1912, the price being $854,500.00, and the time of construction twenty-four months.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	300–00
on L.W.L., feet and inches	300–04
overall, feet and inches	305–03
Breadth, extreme, over guards, feet and inches	31–01
on L.W.L., feet and inches	30–03
Depth, molded, at side to main deck, feet and inches	17–01
Draught to L.W.L., feet and inches	9–05½
Mean trial displacement, tons	1,050
Ratio, length to beam	9.93
beam to length	.1007
Ton per inch immersion at L.W.L.	14.5
Area midship section, square feet	198
L.W.L. plane, square feet	6,040
wetted surface, square feet	9,420
Coefficient of fineness, block	.4276
midship section	.6924
L.W.L. plane	.6648
cylindrical	.6180



U. S. S. "CUSHING."

Capacity of fuel-oil tanks, tons	346.6
reserve-feed tanks, F.W., tons	18.6
fresh-water tanks, F.W., tons	14.6
cofferdam A–16, F.W., tons	18.3
trimming tank A–1, S.W., tons	16.15
D–8, S.W., tons	9.2

The arrangement of the decks, quarters, machinery spaces, storerooms, oil-storage tanks, etc., is in general the same as for other destroyers of this class, Nos. 51 to 56.

MACHINERY.

The propelling machinery consists of Curtis turbines arranged on two shafts as shown on Plate I. On each shaft there is a 30-inch pitch diameter geared cruising turbine connected to the main shaft by means of a Metten clutch, and a 63-inch pitch diameter ahead and astern turbine in a common casing.

The main turbines are designed to develop about 8,000 S.H.P. each when making about 550 revolutions per minute. The cruising turbines are designed to develop about 450 S.H.P. each at 2,000 revolutions per minute and are geared down in the ratio of 7.95 to 1. The cruising units may be used at all speeds ahead below about 16 knots and are disconnected for higher speeds ahead and while reversing.

The main ahead turbine contains forty-one stages, the first four having three moving rows of blades each, and the other thirty-seven being single-bucket drum stages. The 4th stage wheel is forged integral with the drum, the other wheels being of the usual built-up construction.

The reverse turbine consists of one stage with four moving rows of blades and a drum with nine single-bucket drum stages and is capable of developing about 50 per cent. of the ahead power.

The general construction of the turbine is the same as for previous destroyers built by the same company, with the exceptions that the casing for the 1st and 2d stages ahead is cast steel instead of cast iron and all blades over 6 inches in length are soldered at the root.

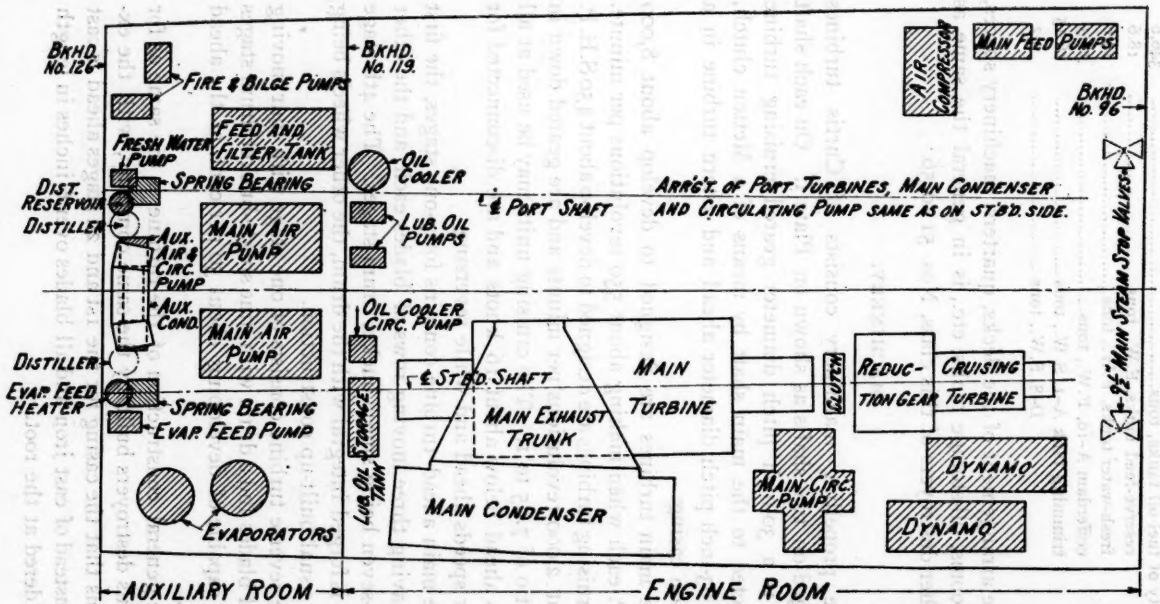


PLATE I – U.S.S. CUSHING – ARRANGEMENT OF ENGINE AND AUXILIARY ROOMS.

All blades, shrouding and channel bars are rolled brass.

On each shaft there is a 30-inch pitch diameter Curtis turbine connected through gearing and a hydraulic clutch to the main shaft. It consists of seven wheel stages, the 1st stage having three rows of moving blades and the other six stages two moving rows each.

The number of nozzles per stage, expansion ratio of nozzles and throat area in square inches is as follows:

Cruising Turbines.

Stage.	Number.	E. R.	Area.	Height, ins.
1	2	1.088	.908	.70
2	10	.....	2.09	.75
3	14	.....	2.926	.75
4	19	.....	3.97	.75
5	27	.....	5.643	.75
6	38	.....	7.942	.75
7	55	.....	11.495	.75

Main Ahead Turbine.

1	20	1.05	10.00	1.0
2	57	.....	22.63	1.125
3	88	.....	34.94	1.125
4	116	.....	46.05	1.125

Reverse.

1	10	1.755	6.75	1.5

SHAFTING AND BEARINGS

Cruising-turbine rotor shaft, length, feet and inches	6–02 9/16
diameter, maximum, inches	06¼
Main rotor and thrust shaft, length, feet and inches	21–06
diameter, maximum, inches	16¼
hole, inches	09
Forward line shaft, length, feet and inches	11–04
diameter, inches	10
hole, inches	07
After line shaft, length, feet and inches	20–08
diameter, inches	10
hole, inches	07
Stern-tube shaft, length, feet and inches	25–08
diameter, inches	10
hole, inches	07

Propeller shaft, length, feet and inches	32–03½
diameter, inches	10
hole, inches	07
Bearings, main, number, each turbine	2
diameter, inches	13
length, inches	18
type	Babbitted.
Spring, number	2
diameter, inches	10½
length, inches	14
type	Babbitted.
Stern-tube, forward, diameter, inches	10¾
length, inches	26½
type	Lignum vitae.
aft, diameter, inches	10¾
length, inches	43½
type	Lignum vitae.
Strut, forward, diameter, inches	10¾
length, inches	31½
type	Lignum vitae.
after, diameter, inches	10¾
length, inches	40½
type	Lignum vitae.

The main thrust bearing is at the forward end of the turbine shaft, and consists of four horseshoes of the regular type, babbitted on both faces and cored for the circulation of cooling water.

The turbine is so designed that the steam thrust balances the propeller thrust at a speed of about 27 knots. At higher speeds the steam thrust is the greater, and at lower speeds, the propeller thrust is the greater. The total effective surface of the horseshoes is 295.2 square inches.

Number of collars	5
Thickness, inches	$1\frac{1}{16}$
Space between collars, inches	3
Outside diameter, inches	14½
Inside diameter, inches	8½

The shaft horsepower developed by the engines is measured by a Gary-Cummings torsionmeter, one for each line of shafting.

The torsionmeter constants were found by calibration of

the actual shafts with the meters mounted on them. The constant for the starboard shaft is 6.79215 and for the port shaft 6.67132.

PROPELLERS.

There is one three-bladed outboard-turning propeller on each shaft. The propellers are of manganese-bronze, cast solid, and the blades are true screw and machined to pitch and polished. The following are the actual propeller measurements:

	Starboard.	Port.
Diameter, feet and inches	7–05¼	7–05$\frac{15}{16}$
Pitch, feet and inches	6–06.152	6–06.011
Ratio, diameter to pitch	.873	.864
Projected area, square feet	26.797	26.78
Helicoidal area, square feet	29.67	29.67
Disc area, square feet	43.69	44.00
Ratio, projected to disc	.6133	.6087
helicoidal to disc	.6791	.6743
projected to helicoidal	.9032	.9026
Lower tip of blade above keel, inches	3¼	3$\frac{13}{16}$
Immersion of upper tip of blade at load draught, ins.	20¼	20$\frac{17}{32}$

MAIN CONDENSERS.

There are two main condensers, pear shaped, of the curved-tube type, the tubes being rolled into the tube sheets. They are built up in the usual manner, the principal dimensions being as follows:

Length between tube sheets, feet and inches	14–00
of tubes as fitted, feet and inches	14–09
Thickness of tube sheets, inches	01½
Tubes, number, each condenser	2,532
diameter, outside, inch	00¾
thickness, B.W.G.	18
Cooling surface, square feet	5,800
Main exhaust nozzle	10 feet, 04 inches by 22 inches.
Diameter, auxiliary exhaust nozzle, inches	07
dynamo exhaust nozzle (starboard only) inches	08
air-pump suction, inches	11
circulating-water inlet and outlet, inches	19½
drain connection, inches	03

MAIN AIR PUMPS.

Each main condenser is provided with a Blake vertical, twin-plex, single-acting beam air pump, located in the auxiliary room, with a steam cylinder 12 inches in diameter and wet and dry-air cylinders 28 inches each, with a common stroke of 18 inches. The wet suction is 9 inches in diameter and the dry 7 inches, both taking off the common 11-inch suction from the condenser. From the wet-air pump there is an 8-inch discharge to the feed and filter tank.

MAIN CIRCULATING PUMPS.

There is one centrifugal circulating pump for each main condenser, driven by a two-cylinder reciprocating engine. The engine's moving parts are completely enclosed and supplied with forced lubrication by a small plunger pump driven from the engine shaft.

Data for One Main Circulating Pump and Engine.

Diameter of impeller, inches	33
suction and discharge nozzles, inches	19½
steam cylinders (2), inches	05½
Stroke, inches	06
Revolutions per minute	300

There is one auxiliary condenser, located in the auxiliary room. It is cylindrical in shape and of the curved-tube type, the tubes being expanded in the tube sheets.

There are 280 ⅝-inch tubes having a total cooling surface of 267 square feet. The length of the tubes as fitted is about 6 feet 9 inches between the tube sheets. Directly beneath and connected to the condenser is a Blake horizontal, simplex, combined air and circulating pump 6 by 8 by 8 by 7 inches.

The auxiliary exhaust connection to the condenser is 6 inches in diameter.

Feed and Filter Tank.—This is located in the auxiliary room and has a total capacity of 818 gallons, of which 235 gallons belongs to the filter chambers and 583 gallons to the

feed tank proper. The tank has a 11-inch connection for main air-pump discharge, a 3-inch connection for the auxiliary air-pump discharge, a 8-inch feed-pump suction connection and a 4-inch overflow pipe. There is also a connection from the distiller fresh-water pump and 2-inch vapor pipes leading from each filter chamber.

FORCED-LUBRICATION SYSTEM.

The main turbine and thrust bearings, cruising-turbine bearings and gears are fitted with a complete system of forced lubrication. The system comprises the following:

1 300-gallon storage tank.
1 200-gallon storage tank.
1 158-gallon oil-settling tank.
1 70-gallon drain tank.
1 23-inch diameter Reilly multicoil oil cooler.
2 4½ by 6 by 6 Blake simplex, vertical oil pumps.

Oil is taken from the drain tank by either of the oil pumps and discharged to the various bearings through the cooler. From the bearings the oil drains by gravity into a 10-inch pipe which runs from the forward bearing of the cruising turbine, aft through the bilge near the center line, to the oil-drain tank.

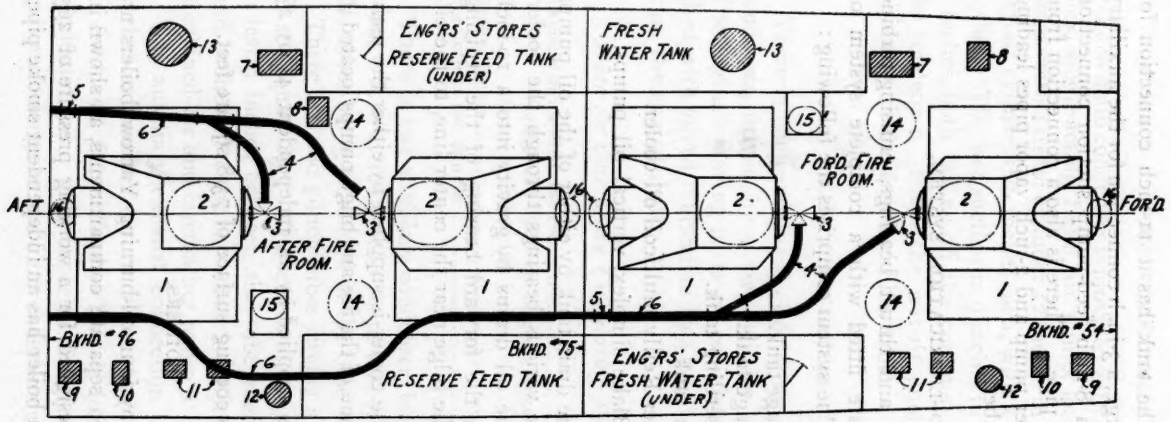
Cooling water for the cooler is supplied by either the main circulating pumps or one of the fire and bilge pumps located in the auxiliary room.

Oil to the clutch is supplied by an independent 4½ by 2¾ by 4-inch pump.

The oil cooler has a cooling surface of 73.8 square feet.

BOILERS.

Steam is supplied by four oil-burning Yarrow boilers arranged in pairs in two separate compartments, as shown in Plate II. They are designed for a working pressure of 265 pounds gage, and each boiler has an independent smoke pipe.



ITEM	DESCRIPTION	ITEM	DESCRIPTION
1	BOILER	9	FUEL OIL HAND PRESSURE PUMP
2	SMOKE PIPE	10	" " BOOSTER PUMP
3	7" MAIN STEAM STOP VALVE	11	" " SERVICE "
4	7½" O.D. MAIN STEAM PIPE	12	" " HEATER
5	MAIN STEAM EXPANSION JOINT	13	FEED WATER HEATER
6	10⅝" O.D. MAIN STEAM PIPE	14	TURBINE DRIVEN BLOWER
7	AUX. FEED PUMP	15	AIR LOCK
8	FIRE AND BILGE PUMP	16	ESCAPE HATCH

PLATE II - U.S.S. CUSHING - ARRANGEMENT OF FIRE ROOMS.

The firerooms are operated under the closed fireroom forced-draft system.

Each boiler is equipped with nine oil burners of the Fore River type.

Data for One Boiler.

Pressure, working, pounds per square inch, gage	265
test, pounds per square inch, gage	400
Drum, steam, inside diameter, inches	43
thickness of tube sheet, inches	1⅛
wrapper sheet, inch	0 9/16
water, inside width, inches	27 1/16
depth, inches	19 23/32
thickness of tube sheet, inches	1¼
wrapper sheet, inch	0 1/32
Number of 1-inch tubes	2,656
1¼-inch tubes	218
Heating surface, square feet	5,375
Furnace volume, cubic feet	630
Diameter of main stop valve, inches	7
feed, stop and check valves, one to each lower drum, inches	2
auxiliary feed, stop and check valve, inches	3
Diameter of surface-blow valve, inches	1¼
bottom-blow valves (two), inches	1½
safety valves (two duplex), inches	4

The auxiliary feed enters the steam drum and the main feed enters the water drums, where it is baffled so that it rises through the two outer rows of tubes.

All tubes are straight except the two rows nearest the furnace, which are curved slightly to provide for expansion. These latter are 1¼ inches external diameter, all others being 1 inch.

FUEL—OIL SYSTEM.

This system consists of two light-service booster pumps, one in each fireroom; four duplex service pumps, two in each fireroom; one oil heater in each fireroom, and the oil-storage tanks, valves and piping as required.

The booster pumps draw oil from the storage tanks and discharge to the suction of the service pumps. The latter draw either from the storage tanks direct or from the dis-

charge line from the booster pumps and discharge to the burner lines.

There are nine burners per boiler, each supplied by a $\frac{3}{8}$-inch branch from the burner line.

The whole system is completely fitted with strainers, stop, and automatic stop valves.

FORCED-DRAFT BLOWERS.

There are two 35-inch single-inlet Sturtevant cone fans in each fireroom, each driven by a Curtis turbine direct connected to the fan. The turbine has one stage with three moving rows of blades.

FEED-WATER HEATERS.

There is one Reilly vertical, multicoil feed-water heater in each fireroom. The heaters are 36 inches in diameter and each contains 29 coils, with a heating surface of 178.35 square feet.

MAIN STEAM PIPING.

The main steam connection on each boiler is 7 inches in diameter, the forward boiler steam lines combine into one $9\frac{1}{2}$-inch line which runs aft on the starboard side. The two after boilers combine into one $9\frac{1}{2}$-inch line which runs aft on the port side, the two main lines being cross connected with a $6\frac{1}{2}$-inch cross connection. There are expansion joints in each line at the engine-room bulkhead and a stop valve at the engine-room bulkhead. Just inboard of the bulkhead stops there are $9\frac{1}{2}$-inch astern and ahead turbine throttles and a 3-inch throttle for the cruising turbines.

AUXILIARY STEAM PIPING.

There is a $3\frac{1}{2}$-inch stop valve on each boiler for supplying the auxiliary steam line. In each fireroom connections are taken off these lines for the various auxiliaries in the fireroom, the two lines in each fireroom then combine into a common

3½-inch line. In the after fireroom these lines are cross connected and a 4-inch line runs aft on each side of the engine room to the auxiliary room, where they are cross connected, forming a loop. From this line branches lead to the various auxiliaries.

AUXILIARY-EXHAUST PIPING.

The auxiliary-exhaust piping runs throughout the machinery spaces, varying in diameter from 2½ to 8 inches. There are branches to each auxiliary and the following connections:

4½ inches to each feed-water heater.
6½ inches to the 1st and 13th stage of each main turbine.
6 inches to the auxiliary condenser.
7 inches to each main condenser.

MAIN AND AUXILIARY FEED SYSTEMS.

There is an 8-inch suction main running forward from the feed and filter tank with a 5½-inch connection to the air-pump channel ways and a 5½-inch connection to each main feed pump in the engine room. The line runs forward to the after fireroom, where it divides into two 5½-inch branches, one to each auxiliary feed pump in the two firerooms.

The discharge from each feed pump is 4 inches in diameter, the discharges from the two main feed pumps combining into one 6-inch line which runs forward to the firerooms, with a 4-inch branch to each feed-water heater or by-pass, thence to the lower drums of the boilers.

Each auxiliary feed pump discharges direct through the auxiliary feed line to the upper drums of the boilers or cross connects to the main line, thence through the heater or by-pass to the lower drums.

EVAPORATING AND DISTILLING PLANT.

There are two evaporators in the auxiliary room and three distillers in the hatch over the auxiliary room. They have a nominal capacity of 3,750 gallons of potable water per 24

hours, and when clean are to have an overload capacity 40 per cent. in excess of the nominal capacity.

Data for One Evaporator.

Type	Reilly, vertical, multicoil.
Diameter, inside, inches	32
Height overall, feet and inches	6–06
Number of coils	10
Diameter of coils, inches	04⅜
tubes, inch	01
Thickness of tubes, B.W.G	16
Heating surface, square feet	61.5
Diameter of steam nozzle, inches	01½
vapor nozzle, inches	02½
feed valve, inch	01
blow valve, inches	01½

Data for One Distiller.

Type	Reilly, vertical, multicoil.
Diameter, inside, inches	13
Height between covers, feet and inches	5–05⅛
Number of coils	3
Diameter of coils, inches	04⅜
tubes, inch	01
Thickness of tubes, B.W.G	16
Cooling surface, square feet	18.5
Diameter of vapor inlet, inches	01½
drain connection, inch	0⅜
circulating-water connections, inches	01½

TRIALS.

The following trials were required by the contract:

(*a*) A progressive trial over a measured-mile course at Rockland, Me., for standardizing the screws, extending from maximum speed (at least 29 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four-hours' duration in the open sea in deep water at the highest speed attainable, the average for the four hours not to be less than 29 knots.

Three fuel-oil and water-consumption trials (*c*), (*d*) and (*f*), of four-hours' duration in the open sea in deep water at average uniform speeds of 24, 15½ and 12 knots respectively,

as nearly as possible. The trials to correspond as nearly as possible to service conditions. The cruising units to be used on trials (*d*) and (*f*).

(*e*) An endurance trial of ten-hours' duration in the open sea at an average speed of 15½ knots, made under the same conditions as trial (*d*) in order to test the reliability of the cruising units. No fuel or water measurements were required on this trial.

(*g*) Trials to determine the ability of the vessel to back satisfactorily at full and cruising speeds, time to bring vessel to dead stop from a speed of 29 knots ahead and the distance reached, also trials to test ability to back when going ahead at about 15 knots with the cruising units in use.

In addition to the above trials a trial of two hours' duration under the same conditions as trial (*d*), except that the main turbines only to be used was required.

Fuel Guarantees.—The contractors guaranteed that the consumption of fuel oil per knot run for all purposes, including that necessary for all auxiliaries in use on the trial, should not exceed 700 pounds at 29 knots, 411 pounds at 24 knots, 150 pounds at 15½ knots and 120 pounds at 12 knots.

The standardization trial was run on May 25, 1915, on the measured-mile course at Rockland, Maine. In all 27 runs were made; run No. 22, being unsatisfactory, was thrown out. From the data obtained on the standardization trial it was found to require 569.7 r.p.m. of the propellers for a speed of 29 knots; 437.9 r.p.m. for 24 knots; 266.7 r.p.m. for 15½ knots and 206.7 r.p.m. for 12 knots.

Table II contains the data from which the curves, Plate III, were plotted.

The endurance run was held on July 22, 1915, the other trials required by the contract were held on the dates noted in Table III, which also gives the data obtained.

TABLE I—PUMPS AND CONNECTIONS—U. S. S. CUSHING.

No.	Pumps.	Size (Ins.)	Type.	Suction Pipes from—		Discharge Pipes to—		Location.
				Ins.		Ins.		
2	Main Air.	12 2 (2) 28 x 18.	Twin, Vertical, Bucket, Single-Acting, "Blake."	11	Condensers.	6	Feed Tank.	In Auxiliary Room.
				1½	Reserve Feed Tanks.			
				1½	Condenser for Sealing Water.			
2	Main Circulating.	33 Dia. Runner, 5½ x 5½ x 6 Engine.	Vertical, Double Engine, Centrifugal Pumps.	19½	Sea.	19½	Main Condenser.	In Engine Room.
				7	Bilge.			
2	Main Feed.	15 x 10 x 16	Vertical, Piston, Double-Acting, Single, "Blake."	5½	Feed Suction Pipe and Res. Feed Tanks Cross-Conn. Pipe.	4	Main Feed Disch.	In Engine Room.
2	Aux. Feed.	Do.	Do.	5½	Feed Suction Pipe and Res. Feed Tanks.	4	Aux. Feed.	One in each Fireroom.
				1½	Hose Connection.	1½	Hose Connection.	
2	Fire and Bilge.	7 x 7 x 12	Do.	4	Sea.	2½	Fire Main.	In Auxiliary Room.
				4	Drainage.	3½	Overboard.	
				1½	Hose Connection.	1½	Distiller Flushing System thru Fire Main.	
						1½	Hose Connection.	
2	Fire and Bilge.	Do.	Do	3½	Sea.	2½	Fire Main.	In #1 Fire Room.
				4	Drainage.	3½	Overboard	
				1½	Hose Connection.	1½	Hose Connection.	
				4	Independent Bilge.			
1	Aux. Air and Circ.	6 x 8 x 8 x 7	Combined, "Blake."	4	Aux. Condenser.	3	Feed Tanks.	In Auxiliary Room.
				4	Sea.	4	Aux. Condenser.	
1	Evaporator Feed.	4½ x 6 x 6	Vertical, Piston, Double-Acting, Single, "Blake."	2	Sea.	1½	Evaporators.	Do.
				1½	Distiller Circ. Water Disch.			
1	Distiller Fresh Water.	3½ x 4 x 4	Do.	1½	Distiller Reservoir Tank.	1½	Res. Feed Tanks.	Do.
						1½	Ship's Tanks.	
						1½	Cofferdam.	
						1½	Main Feed Tank.	
2	Lubricating Oil.	4½ x 6 x 6	Do.	3	Lubricating Oil Tank.	2½	Bearings.	In Engine Room.
						1	Oil Settling Tank.	
4	Fuel-Oil Service.	6 x 3½ x 8	Vertical, Piston, Double-Acting, Duplex, "Blake."	2½	Oil Storage Tanks.	2	Burners.	Two in each Fireroom
				2½	Booster-Pump Discharge.			
2	Fuel-Oil Booster	4½ x 6 x 6	Vertical, Piston, Double-Acting, Single, Light Service, "Blake."	3	Storage Tanks.	2½	Service-Pump Supply.	One in each Fireroom.
				2½	Deck-Hose Conn.	2½	Storage Tanks.	
						2½	Deck Hose Conn.	
2	Turbine Drain.	3½ x 4 x 4	Horizontal, Piston, Double-Acting, Single, "Blake."	1½	Turbines.	1½	Filter Tank.	One under each Main Turbine.
1	Oil to Clutch.	4½ x 2¾ x 4	Vertical, Piston, Double-Acting, Duplex, "Blake."	1	Tank.	¼	Clutch.	In Engine Room.

TABLE II - U. S. S. CUSHING
STANDARDIZATION TRIAL DATA - MAY 25, 1916.

No. of Run	Time on Course		Speed in Knots	R.P.M.			S.H.P.		
	Mins.	Secs.		Star. Engine	Port Engine	Mean	Star. Engine	Port Engine	Total
1	7	12.7	8.32	141.32	148.00	144.66	96	148	244
2	7	30.9	7.98	137.16	143.88	140.52	121	166	287
3	6	50.7	8.77	148.48	154.97	151.73	151	176	327
Mean of Group			8.26			144.36			286
4	5	0.5	11.98	205.09	211.67	208.38	348	410	758
5	4	54.3	12.23	204.80	209.32	207.06	334	363	697
6	5	9.2	11.64	203.43	208.68	206.06	359	390	749
Mean of Group			12.02			207.14			725
7	3	53.2	15.44	257.91	262.89	260.40	613	737	1350
8	4	2.3	14.86	260.86	264.35	262.61	691	776	1467
9	3	48.1	15.78	261.50	265.64	263.57	657	762	1419
Mean of Group			15.24			262.30			1426
10	3	4.0	19.57	352.40	348.55	350.48	1747	1674	3421
11	2	54.2	20.67	353.34	349.36	351.35	1752	1701	3453
12	3	3.2	19.65	354.92	352.19	353.55	1784	1715	3499
Mean of Group			20.14			351.68			3457
13	2	25.3	24.78	437.88	440.49	439.19	3480	3673	7153
14	2	36.1	23.06	434.72	437.45	436.09	3455	3531	6986
15	2	24.3	24.95	437.03	436.57	436.80	3503	3466	6969
Mean of Group			23.96			437.04			7024
16	2	24.5	24.91	480.21	480.11	480.16	4831	4835	9666
17	2	13.8	26.91	480.56	483.27	481.92	4865	4965	9828
18	2	23.2	25.14	481.64	486.06	483.85	4874	4994	9868
Mean of Group			25.97			481.96			9798
19	1	59.4	30.15	571.94	577.06	574.50	7653	7892	15545
20	2	8.4	28.04	570.97	574.31	572.64	7640	7778	15418
21	2	1.0	29.75	568.16	569.74	568.95	7279	7356	14635
Mean of Group			29.00			569.68			15254
23	1	58.8	30.30	586.65	581.81	584.23	8049	7802	15851
24	2	4.1	29.01	599.64	593.89	596.77	8309	8201	16510
25	1	57.7	30.59	598.07	597.46	597.77	8370	8251	16621
26	2	4.5	28.92	602.36	595.96	599.16	8469	8310	16779
27	1	59.7	30.08	578.18	579.96	579.07	7776	7622	15398
Mean of Group			29.68			593.84			16384

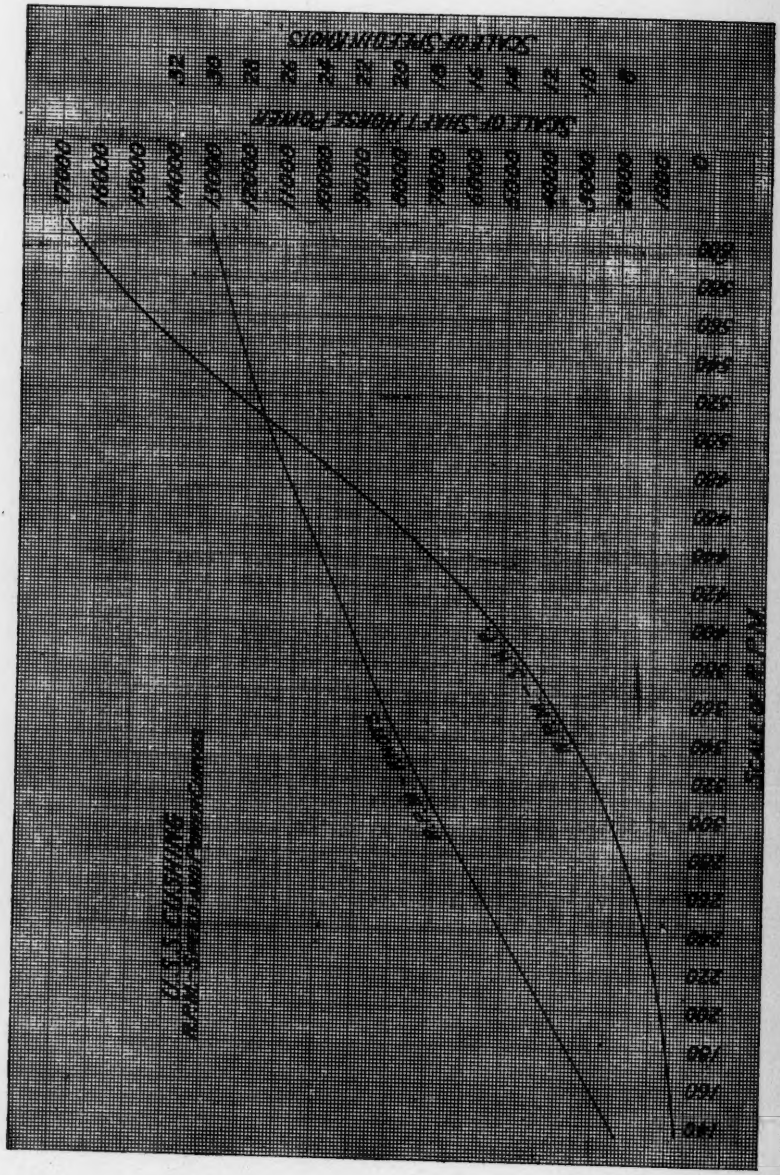


PLATE III.—R.P.M.-SPEED AND POWER CURVES.

PLATE III.—R.P.M.-SPEED AND POWER CURVES.

TABLE III. TRIAL DATA U. S. S. CUSHING.

	4-Hr.Full Speed Trial		4-Hr.24-Knot Trial		4-Hr.15½-Knot Trial		2-Hr.15½-Knot Trial		4-Hr.12-Knot Trial	
Date of Trial	May 26,1915		May 25,1915		June 7,1915		June 7,1915		June 7,1915	
Speed in Knots	29.183		24.178		15.484		15.556		11.988	
Draught, Mean, Feet and Inches	9 - 5		9 - 6		9 - 6		9 - 6		9 - 6	
Displacement (Corresponding), Tons	1048.0		1054.9		1047.6		1050.4		1046.8	
Number of Boilers Used	4		4		2		2		1	
Burners (9 per Boiler)	36		17.375		6		8		4.38	
Nozzles, Each Main Turbine	20		10				5			
Heating Surface Used, Square Feet	21,500		21,500		10,750		10,750		5,375	
per S.H.P.*	1.38		2.95		7.19		7.072		7.466	
Cooling Surface, Sq.Ft. per S.H.P.*	.745		1.591		7.759		7.632		16.111	
PRESSURES, LBS. PER SQ.IN. (AVERAGE)	Star.	Port	Star.	Port	Star.	Port	Star.	Port	Star.	Port
Main Steam at Boilers, Gauge	246.3		238.88		254.51		222.25		222.75	
Engine-Rooms "	224.63	225.13	238.38	237.63	249.0	249.0	210.0	210.0	215.0	215.0
Cruis.Turb. Steam Chest Abs.	---	---	---	---	253.75	253.75	---	---	135.0	135.0
1st Stage "	---	---	---	---	78.75	81.25	---	---	40.56	45.0
Main Turb. Steam Chest "	224.44	226.38	231.88	229.25	---	---	142.5	142.5	---	---
1st Stage "	89.38	90.06	44.75	50.75	11.31	11.06	14.38	14.5	8.56	8.56
3rd Stage "	49.06	47.69	24.25	24.50	5.94	5.69	8.5	8.5	4.0	3.54
Drum Head "	36.19	34.88	17.38	17.19	3.69	2.13	6.06	5.0	2.5	1.38
Aux.Exh.13th Stg."	23.31	22.44	11.44	10.25	3.56	3.0	4.0	3.69	2.5	1.5
Vacuum, Main Cond., Inches of Hg.	27.50	27.54	28.03	28.09	28.38	28.25	28.0	28.0	28.06	28.25
Exh.Shell " " "	25.97	--	26.0	---	26.94	---	26.53	---	27.0	---
Barometer, Inches of Hg.	29.65		29.93		29.90		29.94		30.00	
Auxiliary Steam, Gauge	245.38		243.13		249.0		210.0		215.0	
Exhaust "	9.5		6.25		2.5		2.76		1.13	
Gland Steam "	6.53		4.00		2.10		1.69		1.91	
Main Feed Discharge "	301.25		298.75		245.63		232.5		222.5	
Forced Lubrication "	26.75		30.75		22.13		23.0		23.75	
Oil to Clutch "	---		---		55.0		---		50.63	
Fuel Oil to Burners "	172.27		156.72		136.25		113.13		155.63	
Air in Firerooms, Inches Water	4.40		2.04		1.48		1.84		1.20	
TEMPERATURES, Degrees, Fahr.										
Outside Air	55.63		54.38		52.13		56.25		57.5	
Engine-Rooms	70.56		72.13		83.00		75.50		79.5	
Auxiliary Room	78.31		76.25		76.88		78.50		82.63	
Firerooms	84.19		87.63		90.75		86.75		90.00	
Main Injection	46.81		49.38		62.88		63.38		52.38	
Outboard Delivery	70.06		74.38		67.88		65.88		65.63	
Main Air Pump Discharge	62.60		63.01		59.88		62.75		61.00	
Feed Water from Heater	147.41		173.44		182.25		179.75		191.75	
Oil to Cooler	107.56		107.88		105.63		93.13		100.75	
from Cooler	80.5		89.88		90.00		77.00		82.75	

Main Injection	48.81	49.35	52.88	53.38	52.38
Outboard Delivery	70.06	74.38	67.88	65.88	65.63
Main Air Pump Discharge	62.60	63.01	59.88	62.75	61.00
Feed Water from Heater	147.41	173.44	182.25	179.75	191.75
Oil to Cooler	107.56	107.88	105.63	93.13	100.75
from Cooler	80.5	89.88	90.00	77.00	82.75
Water from Cooler	52.25	64.13	76.75	66.50	64.50
Oil Drains, Cruis. Turbs., Forward	---	---	121.0	---	112.88
Aft	---	---	115.98	---	108.38
Main Turbs., Forward	120.97	116.60	103.32	90.53	97.75
Aft	99.85	100.72	102.29	96.57	96.22
Fuel Oil to Heaters	65.15	66.38	64.00	62.50	67.38
from Heaters	135.00	149.35	114.36	113.25	110.75
R.P.M. Forced Draft Blowers	1674.7	1140.0	1067.5	1381.3	921.25
DOUBLE STROKES PER MIN. OF PUMPS					
Main Circulating	246.2	83.44	110.88	154.44	114.26
Main Air	37.25	30.88	43.75	32.00	36.13
Main Feed	31.38	34.25	---	---	---
Auxiliary Feed	---	---	10.06	13.48	6.87
Fire and Bilge	47.31	38.0	30.75	24.25	31.50
Forced Lubrication	25.88	25.63	39.50	37.75	45.50
Fuel Oil Service	22.25	11.38	4.89	6.61	3.49
R.P.M. Starboard Shaft	578.55	441.35	265.07	265.80	206.12
Port Shaft	573.56	442.14	267.82	269.59	206.64
Mean	576.06	441.75	266.45	267.70	206.38
Slip of Propeller, Starboard	21.52	14.77	9.12	8.95	9.51
Port	20.69	14.76	9.88	10.06	9.57
Mean	21.11	14.77	9.50	9.51	9.54
S.H.P. Starboard Shaft	7768	35.78	550	709	324
Port Shaft	7512	3448	823	850	407
Total	15280	7026	1373	1559	731
from Curve	15675	7290	1495	1520	720
WATER CONSUMPTION, All Machinery					
Lbs.per Hour, Measured	233950.4	125156.2	25869.8	43648.1	13575.5
Reserve Feed	9171.0	2918.0	4790.0	None	5831.0
Evaporated	243121.4	128074.2	30659.8	43648.1	19406.5
" per Sq.Ft. H.S.	11.31	5.96	2.85	4.06	3.61
Lb.of Oil	12.46	13.66	13.20	13.20	13.57
per S.H.P.	15.61	17.57	20.51	28.72	26.95
Knot Run	8331	5298	1980	2806	1619
OIL CONSUMPTION					
Lbs.per Hour	19613.6	9379.2	2325.4	3306.4	1429.8
per S.H.P.	1.259	1.287	1.554	2.175	1.986
Sq.Ft. of H.S.	.9120	.4362	.2161	.3076	.2660
Knot Run	672.1	387.9	150.0	212.5	119.3
Specific Gravity at 60°Fahr.	.9213	.9213	.9135	.9135	.9135
B.T.U. per Pound	19298	19298	19226	19226	19226

*Note.- S.H.P. used is that from Standardization Curve.

THE RESERVE FORCES OF NAVAL MATERIAL.

COÖPERATION BETWEEN THE NAVY AND THE PRODUCERS OF NAVAL MATERIAL.

By H. C. Dinger, Lieutenant-Commander, U. S. N., Member.

The reserve forces of our naval material are represented in the manufacturing plants that produce the material from which the ships, guns and machinery are built and also the commercial shipping which may be converted into naval auxiliaries.

A country without these reserve forces well developed, well founded, prosperous and progressive is at an incalculable disadvantage in comparison with one that, if need be, can put a second fleet into being.

The personnel connected with the production of naval material is also a connecting link between the naval service and the great body of the people. In developing our capacity to manufacture and produce naval material, our naval power is strengthened and developed and is placed on a real sound foundation. A country incapable of building its own vessels and manufacturing its own naval and military equipment is deficient in naval resources and is bound to be at a disadvantage to one that is capable of sustaining and reproducing its own fleet.

It is, therefore, essential that in considering the development of our naval forces the efficiency and well being of the naval reserve of material receive not only attention, but a fostering care.

The ability to build ships and machinery, manufacture guns and armor, is an asset to our naval strength, no matter whether this is in civilian hands or under direct control of

the Navy Department. The commercial dry dock capable of taking the largest naval vessel is as much real assistance to the Navy in time of war as is one at a navy yard. A commercial repair plant that can rehabilitate a damaged dreadnaught, or an ordnance shop that can replace damaged armament, is no small factor in enhancing naval power.

From the standpoint of efficiency and economy, the manufacturing done by the Government should be an adjunct to the manufacturing facilities of the commercial plants of the country. The commercial plant designed to produce naval material can, to a large extent, be utilized also for other than naval work, hence such a plant may bring some return for its cost should naval work fall behind. To increase the facilities for manufacturing naval material much beyond the demands of the Navy building program is a serious economical mistake. It makes no great or material difference to the country at large whether the waste of its manufacturing plants is made by the Government or by the private firm. If there is over production of these facilities, a portion of the plants will lie in idleness and the country loses that much. If the plants are in private hands, they may be utilized to some extent for commercial purposes, but there is still the economic loss.

The Government plant has no reason to be opposed to the private plant, and the best final result both to the taxpayers and to the personnel interested in the manufacturing will be produced when Government establishments and the various commercial plants producing military material operate as properly adjusted component parts of the naval engineering facilities of the country. An over supply of these facilities, which might engender ruinous competition and serious antagonism among the different firms, is of no real benefit to the country, any more than is a ruinous railroad rate war. In a ruinous competition it is very often the labor of the organizations interested that suffers. The loss must be made up somehow and someone must pay for it. The best conditions will, therefore, be produced when there is an adequate amount of naval work to keep all of the plants equipped for this

purpose provided with sufficient work to keep them reasonably employed.

To secure progress, improvement, proper and healthy competition, there should be several firms in the field for any class of work. A monopoly is bad for both progress and cost.

The ideal division of work is such as will keep all of the plants at hand reasonably supplied with work. The total facilities provided, whether under Government or private control, should be proportioned to handle the work. If a definite military and naval building policy were present, the facilities might be properly adjusted so that the plants of the country would have enough but not too much capacity and that they might plan ahead so that they would be most efficient in their equipment and organization. Under these conditions the product could be produced for the least actual cost to the people of the country, and under such conditions our manufacturers would be in the best possible position to cope with foreign competition, and work for foreign governments might then be done with adequate return and as a desirable business venture. To have one plant take work away from another by reason of a slight difference in cost is of no benefit to the people at large, and to have half of the plants idle is decidedly bad for the country. To have all the plants reasonably occupied at a reasonable price for the product is the most desirable result.

The Government can, of course, do a great deal to produce the most satisfactory conditions, but it can not do everything. The governing influences of the various private firms must be willing to coöperate not only with the Government but with each other. The shipbuilder must coöperate with the gunmaker, the steel mill and the founder. It is desirable to have each of the subdivisions of the activities that produce naval material highly specialized in order that the most efficient plant and labor for each part may be present.

The producers of naval material and the Government have a common ground on which to stand and can have a common end in view. There should be healthy competition between

rival firms, but this should not be antagonism. The ability, brain and initiative of the engineering talent should be exercised to develop a better and fitter product, this being the surest means for beating a competitor.

The activities of the Government representatives should be to stimulate these conditions of coöperation, mutual understanding and the furnishing of all possible incentive to the production of the most improved, fit and reliable product that the brains of the American engineer can design and the handicraft of our artisans fabricate. The Navy of this country has developed a field for the inventive genius of the country's engineering talent. It is the largest single customer of the engineering trade. The standards of engineering performance are a guide often used by others. Full, free and open coöperation and a recognition of the mutual interests that exist between the producer and his customer make for better satisfaction to both.

The Navy relies on the commercial engineering field for the excellence of the products from which the material matters of our naval force are constructed. It relies on it for the development of tools, methods of work and the training of artizans by which our fighting weapons are produced in superior form and efficiency. Without a high state of engineering ability and progress in the country at large, the highest character of excellence in navy material can not be realized. The capacity of our commercial engineering plants is the principal asset of our naval engineering reserve of material.

The promotion among the civilian forces engaged in the production of naval material of a feeling of coöperation, a community of interest and a general feeling of fealty to the end in view—that is, the production of the best possible Navy on the appropriation provided by Congress—is a field for development in which the naval service and all those connected with the supply and manufacture of naval material should be particularly concerned.

Harmonious relations between the naval official and those

in the commercial world supplying naval material are essential to the production of the best naval material, and should be one of the essential policies of the Government. The military forces of the Government are for the service of the public, and should be produced in the greatest possible volume for the effort expended. The commercial plants producing naval material are an adjunct to our military preparedness, and should receive such care and treatment by the Government as will put them in the best condition to assist the Navy.

The business relations existing between the Navy Department and the producers of naval material and supplies are an important matter affecting naval administration, and too often this fact is lightly regarded by those of the Navy, by our legislators, by the manufacturers concerned and by the public at large.

A better understanding beween the customer and purchaser whereby a better knowledge of requirements and a better coöperation is arrived at is bound to bring about better conditions and advantage to both.

It is extremely desirable that any prejudice on the part of manufacturers against the Navy work be overcome. Some believe that such prejudice exists, and by reason of it and by reason of some of the conditions said to exist, the Government is not obtaining the most efficient service that might be secured from the sums expended.

Improvement in this field will, therefore, operate as an important means for securing economy and higher efficiency in our naval expenditures.

The two most important matters to aid in improvement of naval business methods are:

(1) Remove all possible causes for unnecessary prejudice, annoyances, delays, misunderstanding, etc., on the part of manufacturers.

(2) Give the best possible publicity and accurate information concerning naval business methods in order that prejudice due to lack of information and understanding may be eliminated as far as possible.

GOVERNMENT PURCHASING BUSINESS.—NECESSARY FORMALITIES AND PROCEDURE.

All Government bodies are governed by laws and regulations, many of them quite undesirable and vexations to the Government officials who have to comply with them and which seem to the Government contractor in many cases a useless mass of red tape. These laws, it may be admitted, are in some cases and, to some extent, ill-advised and make Government business, to some extent, more dilatory and cumbersome than would be necessary in private business.

As long as the laws exist they have to be complied with, and the manufacturers and business men interested in Government work have a far greater opportunity to remedy these laws than have the Government officials who are often blamed and ridiculed for delighting in red tape which they would gladly cut adrift from should the law permit.

Most of the laws, however, have (or have had) a reason for their being, though sometimes the reason may not be deemed meritorious from a business point of view, as, for instance, the eight-hour law, the law that contracts are not to be let to firms engaged in restraint of trade, etc.

It must be recognized at the outset that a Government Department can not do business like a private business man. If this fact is recognized, it may also be accepted as a fact that it is practically impossible for a Government (owing to the nature of its organization and the various restrictions placed upon its discretion in action) to do and conduct business in as efficient and expeditious a manner as can a private business properly organized and administered. A Government Department, other things being equal, is initially handicapped, and on account of the conditions prevaling can not be as economically managed as could a private firm, with the identical personnel. However, by efficient system and wise administration better conditions may be secured and, above all, better coöperation with the producers can be effected. Also by proper effort, bad laws and restrictions can from time to time be removed or modified.

The aim of the Government should be to award contracts in a manner that will secure the most efficient and suitable material at a profit to the manufacturer that will enable him to give (1) fair and just working conditions to his employees, (2) a stimulus to improve his product and (3) to deliver the best possible work, not merely work that can just be slipped through. To do this, unrestricted, free and antagonistic competition is not the remedy, but, on the contrary, is subversive to it. Free competition carried to its logical end naturally results in a cheapening and a deterioration of the product and, from a business and economic point of view, is illogical and inefficient. It is unfair to the people at large and benefits no one except, possibly, some speculating contractors. To require that Government contracts should go without reserve to the lowest bidder is very bad. It is an easy way to let the contract, but it is about the most inefficient one that readily suggests itself (barring favoritism and dishonesty).

For free and unregulated competition there should be substituted a restricted and discretionary competition which would allow the Government to properly make use of other desirable considerations besides the lowest price. This should be arranged so that the responsible Government official might more nearly conduct the making of purchases along the lines used by the reliable and efficient private firm.

This can to a great extent be done in accordance with existing laws, wherever the responsible Government official is willing to take the responsibility in an endeavor to get the best results and not merely follow the beaten pathway of easy precedent.

Restricted competition would consider (1) that the contractor is a bonafide manufacturer or his direct representative; (2) that the contractor has the proper financial backing, reputation, plant and experience that will insure his being able to fulfill his contract; (3) that the official authority is empowered with a certain amount of discretion which takes into account the relative quality, suitability and adaptability of

the product; (4) for special material, limited to naval uses, work should be divided to some extent among at least two firms in order that the facilities of these plants may be kept in efficient condition and enable competition as to price and improvement of quality to be developed. An equitable manner of doing this is to give a larger share to the lowest bidder and a smaller share to others at the same price, or to limit the amount that may be supplied by one firm. The above should be done for such things as aviation equipment, armor, armament, ammunition and special instruments of type designed especially for naval use.

There should be a building up of ability and facilities to compete for business as well as the use of competition to obtain reasonable and fair prices.

New Contractors.—A new contractor is very likely to fall into the fatal error of trying to deal with the Government in the same manner as he has been accustomed to do with an old customer down the street.

Instructions may be sent and clearly stated as to how business is to be done, but these are often disregarded.

A common fault is for contractors to disregard the local inspector to whom they have been advised to address all communications and write direct to the Navy Department. These communications, after sifting through various offices in the Department, are referred back to the inspector who, in most cases, could have furnished the information immediately by telephone.

Large bodies move slowly, and it is an impossibility for an organization as large as the Navy Department to act on matters with as much dispatch as a firm doing, perhaps, a very specialized business. Especially is this so since the Navy Department is, and has for some time been, woefully under-manned to properly handle, according to commercial business ways, the large volume of work required. A private concern doing about forty millions of purchasing a year would have a specially trained and expert force to carefully handle all the details, but the Navy Department is forced to get

along as best it can on the very limited force which Congress grudgingly supplies. It is saving at the spigot and wasting at the bung-hole, but these are the conditions that our legislators have imposed and we have to make the most of them. To materially improve conditions a more specially trained force is necessary.

Contractors' Unfamiliarity.—A great deal of the trouble is due to dealers and manufacturers being unfamiliar with what is wanted and the methods of doing business. The Navy is one of the largest business concerns in the country, and largely by law is compelled to do business in a certain way, which way is designed to be the best that the laws, conditions and the efficiency and capacity of its personnel will allow. Steps are constantly being taken to improve the system, but in many cases it is cumbersome and apparently complicated. This system can not, however, be changed to suit each new bidder that comes along. Among small firms direct contact and communication between the responsible head of the firm is readily made and verbal decisions can be reached. In such a large organization as the Navy, direct contact with the responsible head is almost impossible, and matters must follow a system of routine, or otherwise there would be utter confusion.

The coöperation between the Navy and the producers of naval material is affected by the manner in which the Navy Department does business with the commercial world, and it is important that it be conducted so that the best possible relations exist. The Navy should handle its requirements to secure its products at as reasonable a cost as possible.

Some objections to the manner with which the Government conducts its dealings are made from time to time. Some of these objections are no doubt well founded, and can be remedied. Others, though objectionable from the point of view of the contractor, can not be removed for various good reasons. A setting forth of some of the reasons that make Government business to some extent undesirable, with suggestions for possible improvement or explanation of reasons for the requirements, may serve to clarify the whole matter.

METHODS OF IMPROVING KNOWLEDGE ON THE PART OF MANUFACTURERS OF NAVAL REQUIREMENTS.

The principal method of improving conditions in this respect is to disseminate information to the producers and provide for having authoritative representatives of the Navy confer with and study the possibilities of the different firms who may be in a position to do work for the Navy.

Considering the amount of work involved, little is done by the Navy Department in placing before the commercial engineering world what its requirements are. It has been the practice generally to wait for the commercial engineer to develop something that he thinks the Navy might want and then allow him to press his argument for its adoption. Sometimes they succeed in having laws passed to put their apparatus in use against the advice of responsible naval officials. This usually is a waste of money. This system works after a fashion, but it would be much better for all concerned if the Navy would make special endeavors to state what it was looking for, what it was desirous of having developed, and what it really needs. If this were done in an efficient manner, much delay and working at cross purposes would be avoided. Desired improvements would be more readily met and unnecessary work on the part of both the Government and the manufacturers would be avoided.

The Navy has in the personnel of its inspection force, scattered over the country, a medium for getting in direct touch with the commercial producers of naval material, and supplying this aggregation of producers with first-hand, accurate and reliable information as to what naval products are wanted and desired, how naval work is handled, and other general information and data concerning it.

The inspection force, if made adequate and efficient, can act as a bureau of information to the possible contractors and as a grand system for collecting information for the Navy Department. It can serve to interest a larger number of manufacturers in any particular field and teach them how they can coöperate with the Navy Department and with other

manufacturers by whom their product may be used to produce some implement for naval use.

At present there are inspection forces under the various bureaus of the Navy Department as follows:

Inspectors of Machinery and *Inspectors of Engineering Material*, under the Bureau of Steam Engineering.

Superintending Constructors and *Inspectors of Hull Material*, under the Bureau of Construction and Repair.

Inspectors of Ordnance, under the Bureau of Ordnance.

Inspectors, under the Bureau of Yards and Docks, though most of the inspection at place of manufacture for work under this Bureau is done by the Inspectors of Engineering Material.

If steps were taken to better coördinate these services and secure a general uniformity of doing business under all of the different bureaus, much good would result. Steps for doing a great deal in this line are underway and improvement is slowly but surely being made, notably the following: General specifications for material supplied under Bureau of Ordnance, Bureau of Construction and Repair and Bureau of Steam Engineering are issued, and all specifications for material and supplies under these bureaus are passed upon by a Board consisting of representatives of all the bureaus concerned. Separate specifications under different bureaus are avoided and reduced. Common and standardized practice as to inspection is being developed.

The inspection service is, however, far from all that it should be. To begin with, there is not a sufficient number of officers and expert assistants assigned to the work, and hence the work of collecting and distributing information can only be done superficially.

More officers and the services of mechanical experts in various fields could well be utilized to advantage, especially in connection with conferences of various engineering societies where adoption of standards and development of uniformity in engineering practice is considered. They can also coöperate with the work of other Government Departments, such as the Bureau of Mines and Bureau of Standards. This

matter is most important, not merely to the Navy, but to the general improvement of the commercial efficiency of the country. A few thousand dollars spent in the activities of this service will save millions by securing better results from our naval material and would be worth even much more in the consequent improvement of the commercial engineering efficiency of the country. Therefore every possible step should be taken to improve the *inspection* and *engineering information* service of the Navy so as to make it the most efficient medium for coördination and coöperation between the Navy and the producers of naval material.

Attention may be especially called to the fact that the improvement of the inspection service is to the great advantage of the manufacturing concerns as well as to the Government. Such an inspection service can also do inspection work for any other of the Government departments at very little extra expense. (This was done in the case of the two large colliers building for the Isthmian Canal Commission.) The expense of this inspection to the Canal Commission was trivial because a system and force suited to the work was at hand.

The Navy now buys more than any other Government department and has a more thoroughly organized and widely distributed inspection service for inspecting engineering supplies and material than any other Government department. This force, if made a little more adequate and a bit more efficient, could, if desired, do most of the engineering factory inspection work for nearly all of the Federal Government departments.

A further use of this inspection service is in connection with the work of the Bureau of Standards, Steamboat Inspection Service, etc., for which services the Navy inspection officers could gather information, make engineering investigations, tests in the field, etc.

The expense incurred for such work for other departments could readily be adjusted by transfer of credits or appropriations, a matter that is now readily done.

SPECIFICATIONS.

Practically all naval contracts are let on specifications; therefore the specification is a very important matter and one concerning which considerable objection and criticism has been raised at times. Many of these criticisms, it must be admitted, are well founded. Specifications are troublesome things, but the better and more clear they are, the more efficiently do they serve their purpose.

Objections to specifications consist chiefly (1) In having indefinite terms that do not clearly specify. (2) In having statements that an article must be satisfactory to some official, instead of stating a definite requirement. (3) In giving insufficient information. (4) In requiring unnecessarily costly or special requirements for services where the ordinary commercial material will fully answer. (5) In requiring unnecessarily elaborate tests. (6) In requiring special and arbitrary sizes just different sufficiently from commercial sizes to make the matter a special job, at correspondingly increased cost as well as annoyance.

The greatest possible care must be taken in drawing up the specifications, and this is a matter for which the Government is responsible. Better specifications secure better material at lower cost, hence it is to the interest of the Government to have as expert a force as possible to draw up its specifications. In this matter there is room for a great deal of improvement, though continual advance is being made. The whole matter is considerably limited by the capacity and expertness of the force provided. The Government could well afford to provide a larger, more expert, and a better organized system of preparing its specifications. The purchases aggregate about forty millions a year. Better specifications could easily save four hundred thousand dollars of this sum at an expense of ten or twenty thousand dollars.

The difficulties and annoyance caused by defective specifications is immeasurably greater than the slight extra investigation or labor required to draw up a proper one. This fact

can not be too much impressed upon those who draw up specifications.

The aim of the specifications should be to secure the most economical material for the purpose intended and not necessarily the highest grade of material that can be made. Specifications can easily be made unnecessarily exacting and may require a costly material where one of much less cost would answer better.

Specifications should avoid special or unusual requirements wherever possible and should aim to call for the best standard commercial practice. Every special requirement is eventually paid for as an extra by the Government. Specifications should not hesitate to call for exacting requirements where such are essential and where the extra cost is well applied, but they should not call for special requirements, special quality, odd sizes and dimensions where these are unnecessary.

Efficient specifications can only be prepared when there is provided an expert force for preparing same. The forces provided often are not as expert as they might be if higher qualifications and better pay were provided for certain expert Government employees and it were better realized that a little extra expenditure for improving the system of writing specifications and making purchases would be saved many times in producing better material at a lower average cost. A great improvement is possible by making periodic bulk contracts for as large a number of articles as possible, and for small amounts in emergencies permit of purchase in the open market without advertising. For the large contract sufficient attention can be paid in preparing the specifications so as to be able to get the most serviceable article at the lowest cost.

The Navy Department now issues a considerable number of leaflet specifications covering apparatus and material. An index to these specifications is issued quarterly by the Bureau of Supplies and Accounts. Also General Specifications for Material, General Hull Specifications and General Specifications for Machinery are issued. All apparatus coming within the scope of these leaflet or general specifications should be

required, for under these and the number and designation of the leaflet or general specifications must be clearly stated. If for any reason some departure from the leaflet specifications is desirable, the leaflet should be referred to and the depature or modification clearly stated.

Though all this is required by instructions in force and indicated by common sense, many requisitions and schedules are prepared which are very defective in that they are too general, leave important matters in doubt and do not give sufficient information to the prospective bidder of what is required. The fact that specifications are loosely drawn causes the manufacturer to make an inexact and indefinite bid, or he may bid on something that he really does not actually intend to supply, thinking that, on account of the loosely worded or defective specifications, he may be able to have his product accepted.

DIFFERENCES AND DIFFICULTIES.—WAIVERS.

It has been suggested that matters of this kind should be settled by a Board of Adjudication. This, in most cases, is entirely impracticable and would involve endless red tape. The details of all requirements should be clearly defined in the specifications and contract. Every effort to have inspections, tests, etc., follow promptly should be made. In special instances an appeal to the Bureau concerned can be made, and for good and sufficient reasons certain requirements may be waived.

In the case of contracts for vessels and for other very large contracts, Boards on Changes are provided.

The different bureaus under which material is purchased aim to be reasonable, and in all cases where a contractor has a bonafide and just claim for consideration a proper hearing is given. The Government can not, however, grant every request for waiver that is made nor can it accept responsibility for contractors' mistakes or their lack of knowledge or lack of ability to carry out the contract. There are also some

contractors who will take advantage of the Government should occasion offer.

Jobbers.—Most difficulties are experienced with jobbers who bid without full knowledge, and at the time of the bid have no definite knowledge where they will secure the apparatus or material for which they bid.

Another source of trouble is the manufacturer who does business on his reputation and who considers such practices as requiring a bond or inspection of his product as an insult to his business reputation.

A frequent excuse in case of rejection for non-compliance with contract is that the material is the same as was previously accepted. This is, of course, no reason at all. If the contractor knew his product did not comply strictly with specifications, this should have been stated in his bid, and proper consideration could have been given to the deviation; but to go ahead with the intention of supplying something that does not conform strictly to the specifications because it has previously been accepted is treading on dangerous ground, and though it may go through in some cases, it is very likely to be held up.

IGNORING LOCAL INSPECTOR.

A matter that sometimes causes more or less delay and extra correspondence is the practice, previously touched upon, of some contractors to ignore the local inspector and take up a question direct with the Bureaus at Washington. Such a course merely delays action, since extremely seldom is any action taken without referring the matter back to the inspector concerned.

PAYMENTS.

The Government system of making payments is not difficult to understand if the rules and regulations concerning same are studied. Payment is made on receipt of evidence of service rendered. The supply and forwarding of this evidence sometimes takes considerable time, and this is where most of the delays are caused. Sometimes, too, the

inspectors or authorities receiving the supplies or material fail to report inspection and receipt promptly. In order to have all proper vouchers which are required by law and by the auditor's rulings, somewhat more paper work is required that in most large commercial concerns. The system of payment at present is about as simple as the conditions will allow, and where delays in payment are made it is due to the fault of either contractors or some of the Government officials, through whom the papers are required to go.

The conditions of the payment are stated in the contract, and contractors in making their bid can suggest such conditions as may appear desirable to them, and *such suggestions are considered.*

Partial payments can usually be provided for in contracts involving large amounts, or where there is a long time in delivery.

If the conditions as to payment are definitely stated in the contract, no difficulty or delay in making of payments is presented, but in some cases this point is not clearly brought out in the contract and sometimes as unnecessary hardship is brought upon a contractor.

This matter will in all cases be clear and plain if the *authority* drawing up the requisition or specification will state how payment is to be made and if the *contractor* in making his bid will call attention to any lack of clearness on this point and state terms of payment desired when his bid is submitted.

Difficulties or hardships as to payments can only result due to ignorance or neglect, either on the part of contractor or some Government official, and is not due to the approved method or system of making naval payments.

INSPECTION.

The system of inspection and severity of it have a potent effect.

The final aim of all inspection is, of course, to insure that the contract requirements are met. The inspection should

accomplish this with the least possible cost or delay, both to the Government and to the contractors.

Inspection is directly unproductive, and for that reason must naturally be limited as much as possible consistent with insuring and ascertaining that all requirements have properly been met.

The system of providing inspectors and their assistants is all important. It is a matter that has not received the full consideration that its importance warrants. It is a special class of work, and officers assigned should be well fitted for it, both by experience and professional training.

Inspection is largely a case of a stitch in time saves nine. Of course, if we want to go ahead on the nine-stitch basis no inspection is necessary.

The following are some inspection difficulties known to have taken place:

Material once Inspected at Point of Manufacture Held up at Point of Delivery.

This happens occasionally and has given contractors, at times, reasonable cause for complaint.

These matters are annoying things to contractors as well as to the Government bureaus concerned and evince a lamentable ignorance on the part of the authorities at the point of delivery of requirements of contracts, the specifications and fair dealing. It is the announced policy of the Department that articles inspected at point of manufacture can not be rejected on matters passed by the inspector at the point of delivery, but may be rejected for other defects. It is also the announced policy that the Government can not legally call for more than the contract requires. This seems to be a very difficult thing for many to comprehend—that the contract does not cover the individual opinions of inspecting officers, but only an accurate and just interpretation of the specifications.

The Government, as well as the contractor, is responsible

for the action of its agents, and a matter definitely passed by an inspector stands as passed unless some special requirement permits of other action, or unless the circumstances are such that the contractor has clearly not delivered what was required.

Delay in Inspection.

This is a troublesome matter and at times difficult to obviate, when caused by reason of an overload of work on the inspection service. It is overcome by giving inspectors power and discretion in waiving inspection, accepting manufacturers' certified test reports, etc. For such delays as are beyond control of contractors, delay in delivery is allowed.

Coöperation between the manufacturers and inspection officers can obviate to a great extent delays in inspection.

Articles on which Inspection at Point of Manufacture was called for Shipped Without Inspection.

In such cases the article is often required to be shipped back at the expense of contractor. This is the contractor's or manufacturer's fault; but the constant recurrence of the practice sometimes leads a Government official to suspect that it is somtimes done to avoid inspection, especially when it is known that the article in question is rather urgently needed. The fact that an article is to be inspected at the place of manufacture is clearly stated in the contract, and shipment without inspection is generally inexcusable.

Delay in Receipt of Approved Drawings or in Obtaining Decisions on Questions of Design.

This may be the fault either of the contractor or of the Government. When the fault of contractor, it is usually due to the fact that drawings submitted are inadequate or inaccurate and not in sufficient detail to obtain a proper knowledge of the apparatus. Delay on the part of the Government in this matter is usually due to inadequate force to handle the matter or to lack of coöperation between different Government

forces or to indecision on the part of Government officials as to action to be taken, change of policy, etc.

Another practice sometimes indulged in is for contractors to supply a set of drawings that comply with requirements, but in actual manufacture to use another set or follow shop practices at variance with drawings. This may expedite things occasionally, but sooner or later will be shown up and all the advantage gained will be lost.

Representatives in Washington.

There is also in some quarters a feeling on the part of manufacturers that in order properly to do business for the Navy a Washington representative is necessary. While a representative at Washington is very desirable for the purpose of obtaining special and first-hand information, of being able better to supply information to the officials taking action in various matters and of explaining difficulties, such Washington representatives are not necessary, and a great amount of business is promptly and successfully done without them.

The idea that a Washington representative can influence the obtaining of business or obtain any special consideration in the waiving of deficiencies is a mistaken one. The best influence in obtaining business is a superior product for a lower price. Considerations as to waivers are given on a careful consideration of actual facts, and the Department is guided almost entirely by the recommendation of the local inspector concerned, who makes his recommendations from actual knowledge of the case in point.

DESCRIPTION AND TRIALS OF U. S. S. TORPEDO-BOAT DESTROYER *O'BRIEN.*

By Lieutenant W. F. Cochrane, U. S. N., Member.

The *O'Brien* is one of six destroyers authorized by an Act of Congress approved August 22, 1912, these vessels being designated as Torpedo-Boat Destroyers Nos. 51 to 56, inclusive.

The contract for building three of these boats, Nos. 51, 52 and 53, named *O'Brien*, *Nicholson* and *Winslow*, was awarded to the Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Pa., and was signed December 7, 1912; the time allowed for completion being 23 months for the first boat, 23½ months for the second, and 24 months for the third.

The designed speed was 29 knots, at about 1,050 tons displacement.

The contract price for each vessel was $842,000.00, of which $502,000.00 was allotted for machinery.

The dimensions of vessel, machinery details, etc., are the same as those given in Journal of the American Society of Naval Engineers, Volume XXVII, No. 2, of May, 1915.

TRIALS.

The contract required—

(*a*) A progressive trial over the measured-mile course at Lewes, Del., for standardizing the screws, extending from maximum speed (at least 29 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 29 knots. The speed

to be determined by the average revolutions of the main shafts, according to the official standardization curve.

(*c*) A fuel-oil and water-consumption trial of four hours' duration in the open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service cruising conditions.

(*d*) A fuel-oil and water-consumption trial of four hours' duration at 15½ knots, under conditions similar to the preceding trial, but with the cruising engines connected and in use.

(*e*) An endurance trial of ten hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engines connected and in use. Fuel oil and water consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engines.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea, with the cruising engines connected and in use, at an average uniform speed of 12 knots, as nearly as possible.

(*g*) In addition to the above enumerated trials the contract was amended to include a two-hours' trial at about 15½ knots, with the main turbines only in use, fuel oil and water consumption to be carefully measured on this trial.

Fuel-Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 692.25 pounds at guaranteed maximum speed, 444.6 pounds at 24 knots, 213 pounds at 15½ knots, and 170 pounds at 12 knots, the consumption of the fuel oil at these speeds to be determined by the Trial Board from a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*), and corrected to a standard of 19,500 B.t.u. per pound of fuel oil.

Standardization Trial (*a*).—This trial was first attempted on December 17, 1914, but was discontinued owing to cracking of hull plating. Data for this is not given.

The next trial was held February 8, 1915, with results in Table I-a.

The next was held April 8, 1915, with results given in Table I-b.

The last and successful trial was held May 6, 1915, with results given in Table I-c.

After this a two-hour trial at full power was made, the results of which are given in Table II, together with results on other two full-power trials, after the above mentioned standardization trials.

The balance of the *O'Brien's* trials were held in February, following the second standardization, and the data is not given, as the vessel was not retried at the lower speeds; the data given in Table IV is an average for all three vessels, taken from curves for the *O'Brien*.

Four-Hour, 12-Knot, Fuel-Oil and Water-Consumption Trial (*f*).—This trial commenced at 11:35 A. M., February 5, 1915, and was completed at 3:35 P. M., the same day. The weather was fair and the trial very successful. For data see Table II.

After the trial the *O'Brien* began the completion of her standardization trials.

Four-Hour Full-Speed Trial (*b*).—This trial began at 11:05 A. M., February 6, 1915, off Five Fathom Bank Lightship, and was completed at 3:05 the same day.

The weather was fair, except during the last of the run it was necessary to run head into the sea to avoid fog. The data obtained on this run is given in Table II.

Four-Hour, 24-Knot, Fuel-Oil and Water-Consumption Trial (*c*).—Following the full-speed trial, the 24-knot trial began at 3:25 P. M., and was completed at 7:25 P. M. The weather was fair, with rifts of fog. The trial was very successful.

Two-Hour, 15½-Knot, Fuel-Oil and Water-Consumption Trial (*g*).—The trial began at 5:10 P. M., February 7, 1915, and was completed at 7:10 P. M., February 7, 1915. The trial was very successful.

Four-Hour, 15½-Knot, Fuel-Oil and Water-Consumption Trial (*d*).—The trial began at 7:50 A. M., February 7, 1915, and was completed at 11:50 A. M., same day. The trial was very successful. The weather was fair.

Ten-Hour, 15½-Knot Endurance Trial (*e*).—This trial began at 12:30 A. M., February 8, 1915, and was completed at 11:05 A. M., the same day. The weather was fair. The trial was very successful, and the reciprocating engines and clutches operated excellently.

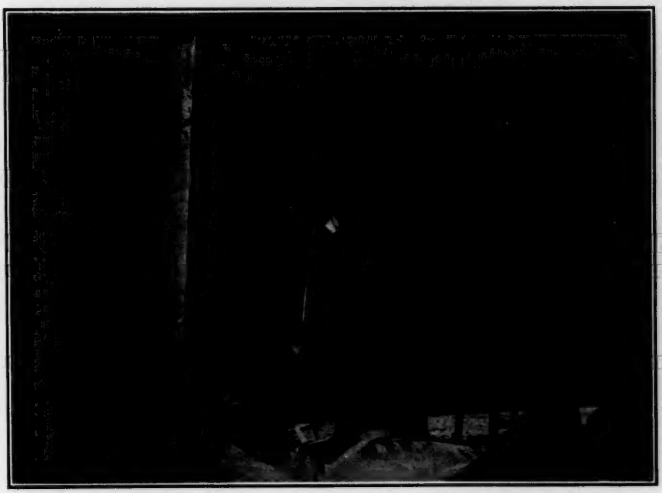
The trials of these three vessels are very interesting in view of the numerous trials conducted with different propellers and with the same propeller in slightly different positions. The *O'Brien* was the ship that made the majority of the various trials, and the performance of all three vessels was practically identical finally, although the *Nicholson's* oil consumptions were better than the *O'Brien's*, and the *Winslow's* considerably better than either of the other destroyers. This was due to vacuum and various other operating features and not to any difference in machinery.

The *O'Brien* began her first official standardization on December 17, 1914. This trial progressed favorably until the 29-knot group of runs began. On the second of three runs the after trimming tanks were reported full of water. This led to an examination. It was found that the hull plating in the wake of the propellers had cracked and the plates bent back. Photos 1 and 2 show the starboard and port sides of the vessel and the openings. The photos were taken after the trial before any repairs had been made. The plates were removed and the heavier plates fitted; stiffeners were also fitted. At the same time the struts were cut down slightly and made very much finer. After three alterations the *O'Brien*, on February 8, 1915, underwent a builders' official trial, the results of which are given in Table II and Curve No. 1. Plate I shows the speed and r.p.m. curve.

On this trial the speed was satisfactory, the oil consumptions were high, although under the guarantees, except at 29 knots. The vibrations of the vessel's hull over the propellers was

U. S. T. B. D. "O'BRIEN"—STARBOARD SIDE.

Forward end of break is vertical on frame 162, about 22 inches high, and extends aft 15 inches on top and 9 inches on bottom. The top is between the 9-foot and 10-foot water lines and the bottom is between the 7-foot and 8-foot water lines. The break apparently started with the vertical rupture at frame 162. The forward end of break is about in line with center of propeller. The distance from hull, at the break, to the propeller tip circle is about 18 inches.



U. S. T. B. D. "O'BRIEN"—PORT SIDE.

Vertical crack through plate on frame 162 at about same height as break on starboard side. The rag shown in the crack was drawn in as the water ran out in the dry dock.

excessive, and the vessel was not accepted. The problem confronting the contractors was an extremely difficult one, and also one about which as many theories as to cause, etc., can and were advanced as there are men to make them, but the only method open to the contractors was one of trial and retrial.

Accordingly a set of spare propellers for the *Aylwin* were obtained from the Government and placed on the U. S. S. *Nicholson.* The measurements of these propellers were as follows:

	Pitch.	Diameter.	Area.	Length of hub.
O'Brien	78 ins.	91½ ins.	3,996 ins.	24¼ ins.
Nicholson	80 ins.	92½ ins.	4,062 ins.	22½ ins.

As can be seen, the propellers of the *Nicholson* were 1 inch greater diameter, 66 square inches larger area, 2 inches greater pitch, and the hub about 2 inches shorter. This brought the leading edge of the propeller 2 inches nearer the strut. With these propellers and some improvements in struts the *Nicholson* ran a very successful trial, the results of which were given in the JOURNAL, Volume XXVII, No. 2, of May, 1915. The vibration was not excessive, the oil consumptions were much better than the *O'Brien's*, and, above all, the *Nicholson* 563.4 r.p.m. for 29 knots, as against 611 r.p.m. for the *O'Brien* at the same speed. The cause of this performance was assigned to various reasons, among them was, difference in hull, + propellers, difference in struts, + propellers, and difference in struts + propellers + position of propellers. To ascertain the cause the contractors measured the hulls and found them the same to all intents and purposes, except the struts on the *Nicholson* were slightly finer.

The *O'Brien* was docked and a new set of propellers installed, having the same pitch, area and diameter as those of the *Nicholson*, but the hub was about 2 inches longer, this making the propeller fit without a distance piece and moving the leading edges of the propeller 2 inches farther away from the strut; also the hull was strengthened by the continuation

of an additional girder up to the main deck and across with full section instead of reduced section. With these alterations a trial was attempted. It was found the r.p.m was slightly less, also the vibrations, but the r.p.m for 29 knots were 593—again high.

The *O'Brien* was docked and the propellers removed and about 2 inches cut off the hub and face and the propellers replaced. This change placed the propellers in same position as the *Nicholson's*, the only difference being one of weight. With these propellers the *O'Brien* ran her final trial with the result that she required more r.p.m. than the *Nicholson*, the vibration was less than the *Nicholson* and the oil consumption on the 2-hour 29-knot run, the only run required, was far better than before.

Various reasons have been assigned as to the causes of this variation ; the best seems to be, that the struts are not exactly in the stream lines, therefore causing broken water to the propellers, and the water close to the leaving edge of the strut is less broken up than a little farther back, and the moving of the propellers forward let them work in a much better area of water.

The performance of machinery in the vessels was par-excellent in all cases. The turbines have a minimum fore-and-aft clearance of $\frac{3}{16}$ inch, and a minimum tip clearance of $\frac{3}{32}$ inch, and are particularly fool proof, requiring no close adjustments, as in the dummy clearances of the Parsons turbines. The blading is extremely strong. It is far stronger than any other marine turbine blading in use at present.

In all respects the turbine is extremely well suited for one or both naval and merchant vessels, where maximum strength, combined with efficiency and ability to stand hard usage without continued watching of fine adjustments that require a very experienced man. The oil consumption of these vessels is higher than the Parsons turbine vessels at high speeds and is lower at low speeds, and is lower than one Parsons vessel at all speeds. Data given in Table IV shows the approximate data for various vessels.

The turbine is properly not a Zoelly turbine, but a Cramp turbine, and as turbines generally receive the name from the engineer of the plant in which they are developed, the credit for the development of this turbine belongs to Mr. J. F. Metten. There is hardly a feature in the turbine today that was in the original Zoelly turbines of the *Mayrant* and *Warrington.*

The tables give the following:

Table I (*a*), Standardization data second trial.
I (*b*), Standardization data third trial.
I (*c*), Standardization data fourth trial.
Table II, Data various trials made by *O'Brien* at same speed with different propellers.
Table III, Data for *O'Brien* and sister vessels at same speeds.
Table IV, Compares *O'Brien*, *Nicholson* and *Winslow* with *McDougal*, *Ericsson and Cushing.*
Plate No. I, Speed curves, Oil curves, S.H.P. curves } *O'Brien.*
Plate No. II, Curves *O'Brien* and sister vessels.

TABLE I (*a*).

STANDARDIZATION DATA U. S. S. *O'BRIEN*, LEWES, DEL., DECEMBER 14, 1914, AND FEBRUARY 5, 6 and 7, 1915.

No. of run.	Speed, knots.	Revolutions per minute.			Shaft horsepower.			Speed, Average group.	R.P.M. Average group.	S.H.P. Average group.
		Stbd.	Port.	Mean.	Stbd.	Port.	Mean.			
1	8.66	134.29	136.40	135.34	.....	.....	264			
2	8.40	148.22	146.96	147.59	.....	.....	295	8.51	143.63	304
3	8.59	142.79	145.21	144.00	.....	.....	360			
4	12.69	205.47	204.87	205.17	.....	.....	749			
5	11.56	204.90	204.69	204.79	.....	.....	820	12.27	204.64	794
6	13.27	204.48	203.12	203.80	.....	.....	785			
7	15.66	263.93	264.97	264.45	731	764	1,495			
8	15.68	263.69	263.34	263.52	796	785	1,581	15.81	263.82	1,547
9	16.22	263.61	263.92	263.77	783	748	1,531			
10	19.41	338.82	338.14	338.48	1,655	1,681	3,336			
11	20.63	339.09	339.17	339.13	1,656	1,669	3,325	20.04	338.82	3,327
12	19.49	338.64	338.46	338.55	1,654	1,666	3,320			
13	24.56	430.64	431.74	431.18	3,598	3,733	7,331			
14	23.27	431.18	432.76	431.97	3,756	4,023	7,779	23.94	431.10	7,596
15	24.64	427.24	431.30	429.27	3,636	3,859	7,495			
16	24.79	494.88	492.77	493.83	5,433	5,757	11,189			
17	27.09	494.29	494.56	494.43	5,351	5,704	11,055	25.93	494.15	11,146
18	24.76	498.72	497.11	497.92	5,475	5,808	11,283			
19	27.13	614.46	608.92	611.69	8,169	8,536	16,705			
20	30.46	605.18	598.28	601.73	8,149	8,076	16,224	28.66	600.16	16,037
21	26.59	589.57	581.42	585.50	7,332	7,660	14,993			
22	29.29	623.91	614.67	619.29	8,451	8,739	17,190			
23	29.41	627.39	615.76	621.58	8,340	8,724	17,064			
24	29.65	637.98	630.82	634.40	8,642	9,220	17,862	29.52	629.27	17,420
25	29.58	638.40	627.45	632.93	8,519	8,921	17,440			
26	29.53	638.99	635.01	637.00	8,527	8,871	17,398			

TABLE I (*b*).

STANDARDIZATION DATA U. S. S. *O'BRIEN*, LEWES, DEL., APRIL 8, 1915.

No. of run.	Speed, knots.	Revolutions per minute.			S.H.P., mean.	Speed, average group.	R.P.M. average group.	S.H.P. average group.
		Starboard.	Port.	Mean.				
1	8.26	139.80	140.06	139.93	258	8.82	140.19	245
2	9.85	139.98	141.39	140.69	246			
3	7.31	139.08	139.77	139.43	230			
4	14.38	199.56	199.91	199.74	730	12.51	199.98	738
5	10.66	200.97	200.21	200.59	742			
6	14.35	198.66	199.33	199.00	737			
7	14.13	258.07	257.32	257.70	1,535	15.99	257.40	1,569
8	17.84	257.66	257.88	257.77	1,600			
9	14.16	256.18	256.85	256.37	1,540			
10	22.51	337.20	338.80	338.00	3,603	20.78	338.64	3,610
11	19.01	337.91	338.57	338.24	3,571			
12	22.58	339.77	340.34	340.06	3,694			
13	22.63	422.71	422.09	422.40	7,697	24.21	421.64	7,644
14	22.68	419.95	421.77	420.86	7,667			
15	22.89	421.74	423.11	422.43	7,546			
16	27.21	479.29	479.51	479.40	10,871	25.97	478.55	10,884
17	24.97	479.64	481.14	480.39	10,964			
18	26.73	472.48	475.54	474.01	10,735			
19	27.91	586.49	589.32	587.91	16,096	28.81	587.11	16,181
20	29.46	584.54	586.60	585.57	16,209			
21	28.41	588.93	589.85	589.39	16,209			
22	30.13	613.61	615.17	614.39	17,830	29.92	621.39	18,091
23	29.58	621.46	620.55	621.01	18,126			
24	30.05	619.20	621.80	620.50	17,982			
25	30.03	625.08	625.76	625.42	18,096			
26	29.85	622.28	623.37	622.83	18,487			

TABLE I (*c*).

STANDARDIZATION DATA U. S. S. *O'BRIEN*, LEWES, DEL., MAY 6, 1915.

No. of run.	Speed, knots.	Revolutions per minute.			S.H.P., mean.	Speed, average group.	Revolutions per minute, ave. group.	Shaft horse-power, ave. group.
		Starboard.	Port.	Mean.				
1	15.14	257.14	255.20	256.17	1,514			
2	16.45	256.00	253.48	254.74	1,479	15.60	254.77	1,473
3	14.34	253.57	253.24	253.41	1,421			
4	21.82	336.72	334.62	335.67	3,377			
5	19.04	339.28	338.27	338.78	3,391	20.45	336.89	3,372
6	21.91	334.04	334.57	334.31	3,329			
7	22.87	430.45	431.41	430.93	7,592			
8	25.97	424.89	425.61	425.25	7,427	24.39	427.14	7,498
9	22.74	427.76	426.46	427.11	7,547			
10	28.06	485.31	484.80	485.06	11,022			
11	24.76	484.98	482.59	483.79	10,728	26.41	484.64	10,899
12	28.06	486.88	484.96	485.92	11,116			
13	27.50	574.11	572.69	573.40	15,011			
14	30.51	575.26	572.08	573.67	15,623	29.03	574.62	16,469
15	27.59	578.54	576.94	577.74	15,618			
16	30.80	586.97	587.82	587.40	16,257			
17	28.37	604.37	604.02	604.20	17,086			
18	30.93	596.99	597.37	597.18	16,737	29.68	599.47	16,751
19	28.48	598.41	600.24	599.33	16,377			
20	31.01	607.16	606.62	606.89	17,346			

TABLE II.—U. S. S. "O'BRIEN;" FOUR-HOUR FULL POWER.

	4 Hours.	1 Hour.	2 Hours.
Date	2/6/15	4/8/15	5/7/15
Displacement	1,052.0	1,053.0	1,053.7
Boilers	4.0	4.0	4.0
Heating surface	21,600.0	21,600.0	21,600.0
Speed, in knots	29.168	29.03	29.053
R.P.M., starboard	618.31	593.49	575.78
port	615.19	593.83	574.97
mean	616.75	593.66	575.38
S.H.P., mean	16,275.0	16,620.0	15,386.0
Pressures:			
Main steam	245.0	251.0	249.0
Full speed (abs.)	215.0	211.0	206.0
Cruising speed (abs.)	170.0	160.0	150.5
Tenth stage (abs.)	28.0	26.5	25.2
Fourteenth stage (abs.)	16.5	15.3	15.7
Gland steam	6.4	5.0	5.0
Auxiliary exhaust	7.2	7.0	6.0
Vacuum, starboard	27.0	29.0	28.5
port	27.5	29.2	28.7
Lubricating oil to M.E.	11.4	11.0	10.1
Fuel oil per knot run	794.24	707.32	643.93
Propeller, pitch, inches	78.0	80.0	80.0
diameter, inches	91½	92½	92½

TABLE III.—FOUR-HOUR, FULL POWER.

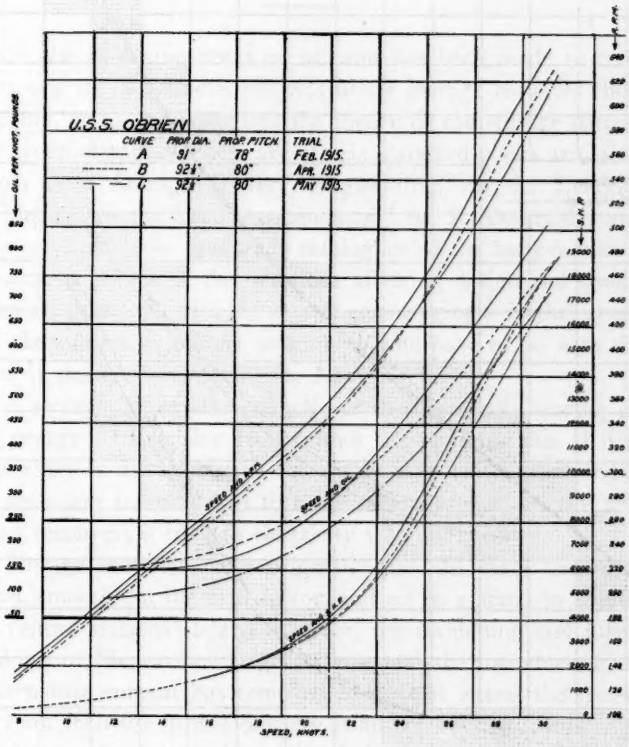
	O'Brien.	*Nicholson.*	*Winslow.*	*Balch.*
Date of trial	5/7/15	3/24/15	6/30/15	2/26/14
Displacement	1,054.7	1,045.0	1,041.0	1,053.9
Boilers in use	4.0	4.0	4.0	4.0
Heating surface	21,600.0	21,600.0	21,600.0	21,600.0
Speed in knots	29.053	29.084	29.054	29.618
R.P.M., starboard	575.78	566.40	573.21	596.96
port	574.97	566.79	573.09	597.17
mean	575.38	566.10	573.15	597.06
S.H.P., mean	15,386.0	15,906.0	15,984.0	17,251.0
Pressures:				
Main steam (G)	249.0	250.8	250.0	248.0
Full speed (abs.)	206.0	208.2	207.0	215.0
Cruising speed (abs.)	150.5	157.5	152.5	200.0
Tenth stage (abs)	25.2	25.2	25.3	31.6
Fourteenth stage (abs.)	15.7	14.3	15.0	.....
Gland steam (G)	5.0	6.3	6.0	8.3
Auxiliary exhaust (G)	6.0	8.0	8.0	6.0
Vacuum, starboard	28.5	27.6	28.0	28.9
port	28.7	27.8	27.8	29.03
Lubricating oil to M.E.	10.1	12.6	10.0	15.7
Fuel oil per knot run at 29 knots	643.93	687.3	661.50	753.54

Propellers same for all vessels.

TABLE IV.—FULL POWER.

	O'Brien, *Nicholson*, *Winslow*. Cramps' turbines.	*McDougal*. Parsons turbines.	*Ericsson*. Parsons turbines.
Displacement, tons	1,045.0	1,021.0	1,087.1
Speed	29.06	30.697	29.29
R.P.M., average	571.0	608.99	616.02
S.H.P., average	15,758	16,974	17,479
Vacuum	.....	.....	.....
Fuel oil per knot run at speed given above	649.5	645.0	746.2
Twenty-four Knots.			
Displacement, tons	1,046.0	1,013.4	1,047.8
Speed	24.081	24.074	24.115
R.P.M.	416.0	417.27	433.32
S.H.P.	7,300	6,413	7,150
Vacuum	.....	.....	.....
Fuel oil per knot run at 24 knots, water	388.3	325.0	392.55
Fifteen and one-half Knots.			
Displacement, tons	1,053.0	1,019.0	.....
Speed	15.59	15.55	15.448
R.P.M	251.0	256.45	258.04
S.H.P.	1,600	1,236	1,099
Vacuum	.....	.....	.....
Fuel oil per knot run at 15½ knots,	138.0	148.0	193.2
Twelve Knots.			
Displacement, tons	1,050.0	1,020.0	1,096.3
Speed	12.08	12.51	12.056
R.P.M	194.2	195.45	198.62
S.H.P.	672	517	435
Vacuum	.....	.....	.....
Fuel oil per knot run at 12 knots.	105.0	129.6	156.3

Plate I.


U.S.S. O'BRIEN
CURVE PROP. DIA. PROP. PITCH TRIAL
A 91" 78" FEB. 1915.
B 92½" 80" APR. 1915
C 92½" 80" MAY 1915.
GAL. PER KNOT POUNDS
S.H.P.
SPEED, KNOTS.

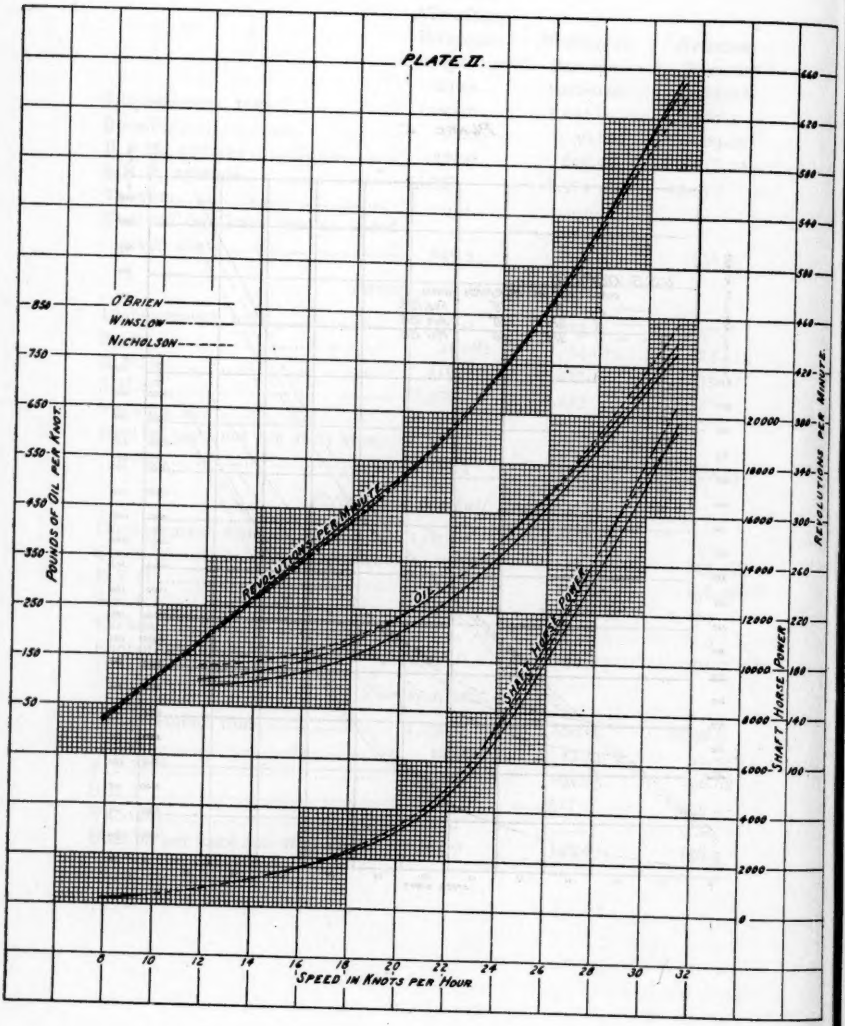


PLATE II.
O'BRIEN
WINSLOW
NICHOLSON
REVOLUTIONS PER MINUTE
OIL
SHAFT HORSE POWER
POUNDS OF OIL PER KNOT
REVOLUTIONS PER MINUTE
SHAFT HORSE POWER
SPEED IN KNOTS PER HOUR
0 10 12 14 16 18 20 22 24 26 28 30 32
850 750 650 550 450 350 250 150 50
460 420 380 340 300 260 220 180 140 100
20000 18000 16000 14000 12000 10000 8000 6000 4000 2000 0

NOTES ON STORAGE BATTERIES.

By Lieutenant C. S. McDowell, U. S. N., Member.

In the following notes no attempt has been made to treat any one of the various subjects under storage batteries thoroughly, or to go deeply into the theory of the storage battery or accumulators, as there are certain standard books available, such as: "Storage Battery Engineering," by L. Lyndon; "Theory of the Lead Accumulator," by F. Dolezalib; and various handbooks and trade catalogues of the battery manufacturers. Some of the questions affecting design and operation are presented with the idea of encouraging a further study of the subject by officers and others who have to do with the use of storage batteries in the Navy.

A storage battery is a vehicle for the chemically storing up of energy. Electricity itself cannot be stored, so that if it is necessary to use it at some time other than when generated it is necessary to convert it to some other form of energy and then reconvert it back to electricity when desired.

Storage batteries were originally used to carry the peak load, smooth out the load factor, and act as a stand-by source in central stations; lately, however, the combining and interlinking of the various large systems and the introduction of alternating current has removed to a great extent the necessity for their use in this capacity at all but isolated plants. Of late years the principal output of the battery manufacturers has been for electric vehicles, train lighting, gas car starting and lighting and submarines. In all the above uses a fairly rugged battery of minimum weight and space is required, so that the later developments in storage-battery engineering have been along those lines, which is fortunate for the Navy, for

we must depend for improvements and developments along the Navy's particular lines upon the commercial development along similar lines.

TYPES.

There are at present two widely different types of storage batteries, the lead acid and alkaline. Each type has certain advantages and certain disadvantages; for certain services one type is preëminent and for other purposes the other is best. Each particular class of storage-battery installation should be carefully studied and worked out to obtain the desired results, and both types of batteries should have their inherent characteristics weighed to bring out the advantages and disadvantages for the particular service to which the installation will be put. Neither type of battery is a cure-all for all evils; thus for central-station stand-by service where a large battery is installed and where there may not be more than three or four cycles a year of charge and discharge, some form of lead battery would naturally be installed on account of cost alone, and in general for stand-by service the lead-paste type would be chosen. On the other hand, where a battery for any service may be expected to have very hard mechanical usage, where it will be discharged practically every day and where it will likely be left standing idle while discharged and in general not receive much attention, such as batteries installed in dories and other small boats for lighting, etc., the nickel-iron alkaline (Edison type) would naturally be chosen over lead type, now in common use, on account of its much greater mechanical strength and ability to stand such service for a long time. In this case the cost per year should be less than with the lead types. Certain batteries are installed for "floating on the line," that is, connected in parallel with the normal power supply, so that when the regular supply fails the battery immediately takes up the load. Such an installation acts as the air chamber on a pump, in that if supply voltage rises, the battery will start charging, taking power from the line, while if the supply voltage falls, the battery will discharge to the

line, in each case tending to keep the line voltage constant. When floating on the line the battery should be in a charged condition, and the normal voltage of the supply should be just at the closed-circuit voltage of the battery and high enough so that the battery, if discharged, will take practically a full charge. In addition to the above the discharge-voltage curve of the battery should be flat enough to keep within the voltage limits necessary for the apparatus served by the line. Figures 1 and 3 show characteristic charge and discharge curves of the Edison type of cell, and Figures 2 and 4 show characteristic charge and discharge curves of a lead cell.

It will be seen that the range in voltage on an Edison cell is from 1 volt discharged to 1.8 end of charge, while on a lead cell it is from 1.8 to 2.65 volts at normal rates. The Edison battery will float on the line and take a partial charge at 1.55 volts per cell, while the lead storage battery will float on the line and charge, if discharged, at 2.3 volts per cell. Thus on what is termed 20-volt service, with the lead battery the generator should give 23 volts (10 lead cells used), and if the generator fails the battery will pick up the load at 20 volts, and the drop will be gradual down to 18 volts, and when the main power supply is cut in again the battery will practically pick up a full charge. Under the same conditions 16 Edison cells might be used, in which case the battery could float on the line with the generator giving 24.8 volts; if the generator failed the battery would maintain the supply from 22.5 volts (approximately) to 16 volts, and when the generator was again on the line the battery would take only a partial charge, the battery would have to be charged separately or the voltage raised. The range in voltage of the Edison battery on discharge would also be greater than the limits required by the usual 20-volt instruments.

The above cases are only given to emphasize the statement that because one type of battery is best for one service is no reason for assuming that it will be best for an entirely different class of service.

ELECTROLYTE.

For any type of battery the question of purity of electrolyte is of the utmost importance. In the Edison battery the electrolyte consists of a 21 per cent. solution of potassium hydrate in pure distilled water with a small percentage of lithium hydrate. To insure pure electrolyte being used the Edison Company require in their guarantee that only electrolyte as furnished by them be used. The Navy purchases this in the powdered form to save the transportation of useless weight of water, but when mixing and when adding water to make up for evaporation, only pure distilled water should be used. The amount of water required for make up in the Edison cells is, as a rule, greater than in the lead, due to the greater electrical losses when charging and thus greater generation of heat and evaporation of water. In lead-cell installations with individual cell ventilation, however, excessive evaporation is obtained unless the blowers are carefully regulated. Even with the use of pure distilled water in the Edison cell the electrolyte becomes gradually converted to potassium carbonate through absorbtion of carbon dioxide, with a consequent reduction of capacity of the cell, so that after about 250 cycles this electrolyte should be renewed. The distilled water used should be freshly distilled and not allowed to become aerated.

In the electrolyte for lead batteries the allowable impurities in the acid are specified, and should not be exceeded. Acid procured in accordance with Navy standard specifications is always well within the allowable limits, and the impurities found in electrolyte in cells which have had considerable use can seldom be traced to the acid. The greatest opportunity for impurities to get in the electrolyte is through the water added to make up for evaporation. As the impurities do not evaporate with the water, very small quantities of iron in the water will cumulate and eventually increase to such a proportion as to be harmful, causing local action, the giving off of hydrogen when standing idle, and the reduction in capacity of the cells. There is a certain amount of impurities in the

lead plates, both positive and negative, a chemical analysis of a number of plates of the paste type shows approximately .06 per cent. of iron present. The impurities in the plates will be gradually absorbed by the electrolyte, and as the total iron in the electrolyte should be kept lower than .01 per cent., it is seen that the iron in the plates may be very injurious. This iron probably gets in to the paste due to the use of iron rollers in grinding the active material, and great care by the manufacturers is required to overcome this. Another probable source of impurities is from the rubber jars and separators, both of which contain a certain percentage of iron. However, as stated previously, the greatest source of danger from impurities is in the added distilled water. The absorptive power of distilled water is second only to that of acids, all but the precious metals and block tin are suspectible to its action, and if it comes in contact with tanks or pipes or other metals, or wood, it will be found to contain slight quantities of them in solution.

The specific gravity of the electrolyte is stated as at a certain temperature and with the cell charged, the Navy have adopted 80 degrees F. as the standard reference temperature. The rate of diffusion of acid into the pores of the plates increases with the temperature, so that the capacity of a cell is markedly increased at higher temperatures.

The recommended gravity of the electrolyte varies with the types of plates, the amount of separation, etc., and it is usually stated by the manufacturer. In Figure 6 curves are given to show the increase in capacity when using higher-gravity electrolyte; 1.250 is usually, however, taken as the maximum, for although the capacity is increased when using stronger electrolyte, the life of the cell is materially shortened, and this should be resorted to only in emergencies when it may be desired to get more than rated capacity out of the cell. The effect of impurities is greatly increased in concentrated electrolyte.

In the lead battery the state of charge may be accurately

obtained from the gravity of the electrolyte provided the gravity when fully charged and at a certain temperature is known. In addition to determining that a battery is fully charged by means of the gravity, it is usual to keep charging until the voltage has remained constant and the battery is gasing freely. In the Edison battery, however, the state of charge cannot be accurately obtained from the gravity on account of the fact that the specific gravity of the displaced material is approximately the same as the electrolyte itself.

CHARGING.

In Figures 1, 2, 3 and 4 two methods of boosting are shown for charging to near full capacity in a minimum of time. Overcharge at high rates is injurious, but the only factors in high-rate charging which act injuriously are gasing and heating, and these occur only when more current is being passed through the battery than the plates can store. Therefore any current rate which the cells will absorb without gasing is not injurious, and it is upon this principle that boosting is applied. The nearer discharged the battery the higher charging current it can take, and by starting the charge at a high rate and tapering to a low rate a large proportion of the discharge can be put back in very short time. Where conditions permit, charging at a constant potential is probably the ideal method since it is automatic and requires little attention. In the case of a lead battery it requires 2.3 volts per cell of the battery, and in the case of the Edison battery approximately 1.7 volts per cell of the battery. In Figure 2 there is shown characteristic curves of constant potential charging of a lead submarine cell. In this case a charge of approximately 80 per cent. full capacity has been put into the battery in two hours and fifteen minutes, and approximately 50 per cent. of its full capacity has been put in within one hour. The curves in Figure 1 show an Edison cell charging in series with the lead cell, the current being the same in both cases but the voltage curve of the Edison cell is not a straight line due to the different in-

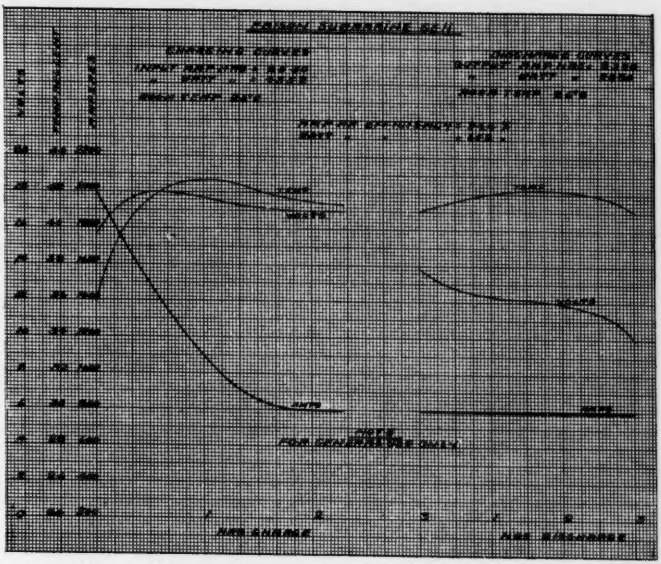


FIG. 1.

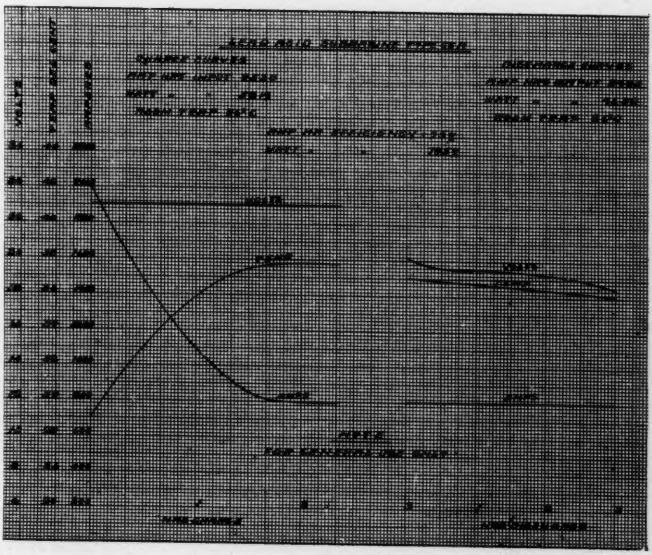


FIG. 2.

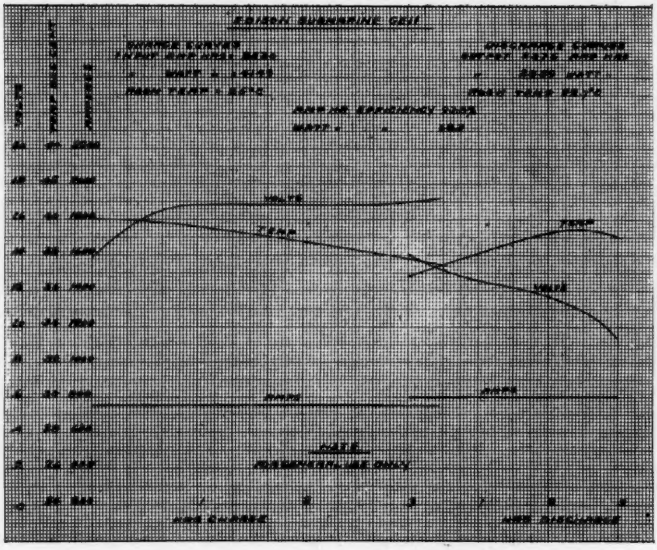


FIG. 3.

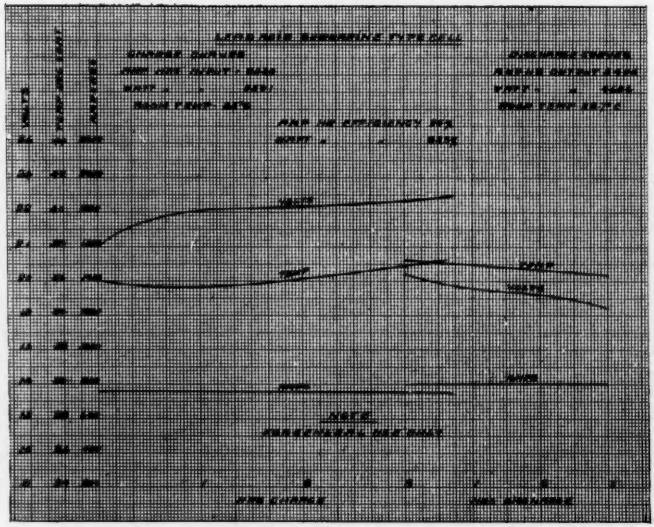


FIG. 4.

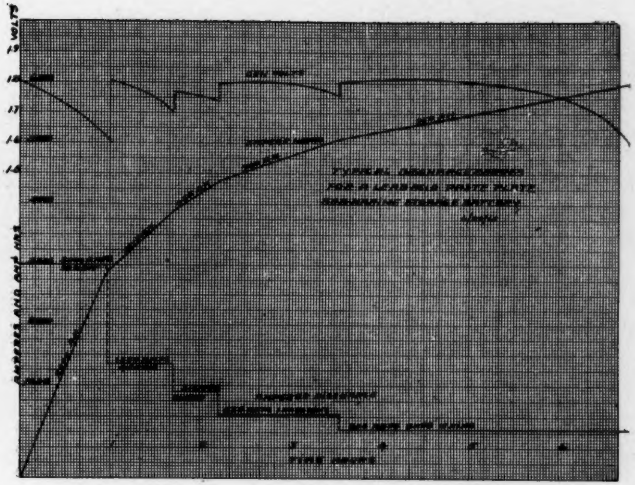


FIG. 5.

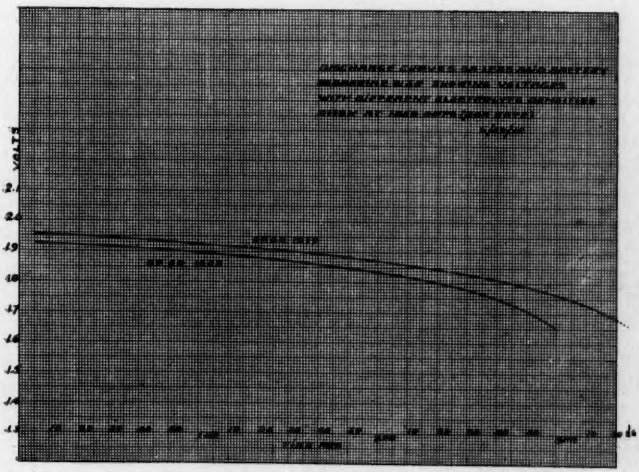


FIG. 6.

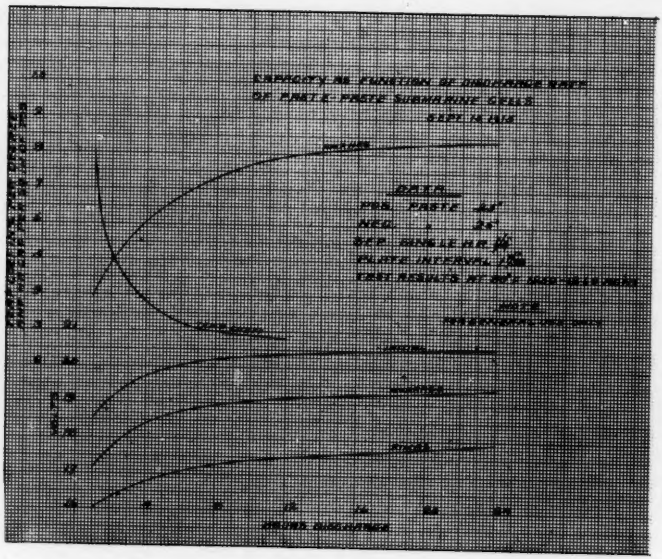


FIG. 7.

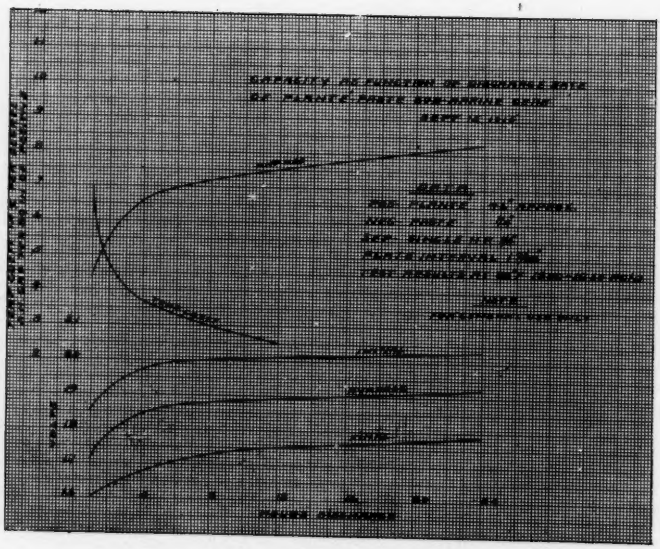


FIG. 8.

ternal characteristics. The limiting temperatures in charging for the lead battery are approximately 44 degrees C. (110 degrees F.), and for the Edison 46 degrees C. (115 degrees F.). It will be noted that in neither case shown by the figures does the temperature approach the above limits.

In the Figures 1 and 2 there are also given the discharge curves of the two types of cells at the 800 ampère rate, it will be noted that the ampère-hour efficiency and watt-hour efficiency in both types is higher than that obtained when the cells are fully charged. In addition to the advantages already given, a battery would apparently have a longer life when worked from 10 per cent. charged to 90 per cent. charged than when worked to the limits in both directions. It is necessary, however, to give periodic overcharges to keep the battery up to full capacity. The principal deterioration of a cell is the shedding of active material which is caused by the gasing and large variations in temperature, and in boosting both of these are minimized.

In certain cases it is more convenient to boost at a constant current on account of the limits of the generator and the high initial ampèrage required by the constant voltage method. To determine what current rate can be safely used for boosting, the following rule may be used:

$$\text{Charging current in ampères} = \frac{\text{Ampère hours discharged.}}{\text{1 plus hours available for charge.}}$$

As an example, if 2,400 ampère hours have been taken from the battery, and one hour is available for charging.

$$\text{Charging current } \frac{2{,}400}{1 \text{ plus } 1} = 1{,}200 \text{ ampères.}$$

The above method will not put in as great a charge in the same time as the constant-potential method, and the current must not be continued beyond the time for which the rate is figured, or gasing and heating will result.

In Figure 5 is shown by curves the recovery of a lead bat-

tery after discharge at high rates. When discharging at the one-hour rate the battery shows low efficiency due principally to the fact that all the energy stored cannot be taken at such a rate on account of the poverty of the electrolyte below the surface of the active material, and the discharge is mostly the surface capacity only. After shifting to a lower rate the acid diffuses in to the inner mass of the active material, and by gradually changing the rate of discharge a total capacity equal to the 20-hour rate is approximately obtained as shown. The 20-hour rate of this particular cell shown is 300 ampères, or a total of 6,000 ampère hours. It should be noted that this total ampère hours has been taken out in six hours and fifty minutes. Discharging in this manner is practically the reversal of the "boosting" method of charging.

In Figures 7 and 8 the effect of the temperature on capacity of paste positive, paste negative lead cells, and Plante positive, paste negative lead cells is shown by curves, and in addition curves are given which show the effect on capacity of the rate of discharge. These curves show the general characteristics only, and should not be applied to any particular case, as the results vary with the material and design of plates, with the method and the amount of separation, and strength of electrolyte.

In order to obtain the full capacity from a battery both the internal resistance and the external resistance should be kept at a minimum, the internal resistance is, in general, controlled by the design, but the external resistance can generally be controlled and kept low by properly looking out for the connections to see that good electrical contact of sufficient area is obtained. It is extremely difficult to keep a good electrical contact when lead connections are bolted together, because lead oxide tends to form, which acts as an insulator; for this reason it is desirable to lead burn all connections wherever possible. As an illustration of the difficulties experienced in bolted connections the case of certain cells may be taken in which the plates of the cell are divided in three groups, the

different groups being connected by bolted equalizing bars. It has been noted that, in a number of cases of such construction, the drop across the connections has been as great as 50 millivolts instead of approximately 2 millivolts which should be obtained. The effect is to throw most of the work on one or two of the groups which, when connected in series with cells of full capacity, will be overcharged and overdischarged, and in cases reversed, giving rise to buckled and cracked plates and short circuits. Some cases have been noted where one or two of the groups of a cell were completely insulated, causing the full capacity to be taken from one or two groups.

SEPARATORS.

The purpose of the separators is to keep the plates from short circuiting by mechanical contact, to keep the separation between plates sufficient so that the required amount of electrolyte may be present between them, to be of sufficient porosity for the diffusion of the electrolyte, and should contain no materials which will injuriously affect the plates. Considerable trouble is experienced in obtaining separators which meet all the requirements. Those principally used are single hard rubber, double rubber, wood, and wood and rubber. In the case of wood separators it is very difficult to get them entirely free of acetic acid and, as a rule, they should not be used against the positive plates; the effect on the negative plates is not so harmful, and when used in conjunction with hard rubber the wood separators should be next to the negative plates. Wooden separators when not in use should be kept submerged in distilled water so as to prevent their drying out and becoming brittle. In the case of single-rubber separators a considerable number of holes are necessary to give sufficient porosity, these holes provide a path for the lead trees found in the electrolyte to bridge the space between plates and cause short circuits; in addition, it is difficult to obtain a design of single-rubber separator which will remain in a plane—

the separators tend to take on a waved surface, forming pockets in which the shedding active material collects and short circuits the plates. Short-circuited plates should be readily detected by the fact that the cells in which they are contained cannot be kept up to capacity and voltage. The best practice at present is to install either double-rubber or wood-and-rubber separators.

DESCRIPTION AND TRIALS OF U. S. S. *FULTON.*

(*Submarine Tender No. 1.*)

By Lieutenant (J. G.) C. N. Hinkamp, U. S. N., Member.

The builders, Fore River Shipbuilding Company, and the contractors, The New London Ship and Engine Company, have recently successfully completed the final trials of the U. S. S. *Fulton.*

Officially known as *Submarine Tender No. 1*, the *Fulton* is the first submarine tender built for the U. S. Navy, and was authorized by an act of Congress, approved March 4, 1911. The contract for her construction was signed June 19, 1912, the time of completion being twenty-four months and the price $492,930.00. She is a single-screw vessel, driven by Diesel engines, connected to the propeller shaft by a jaw clutch, and designed for a speed of 12.25 knots at a displacement of 1,407 tons, with the engines developing about 1,000 horsepower.

PRINCIPAL HULL DIMENSIONS.

Length on L.W.L., feet and inches,	216–00
Length, over all, feet and inches,	226–06
Beam on L. W. L., feet and inches,	35–01.3125
Beam, extreme, feet and inches,	35–01.3125
Draught to L.W.L., mean, feet and inches,	12–11.5
Displacement corresponding, tons,	1,407.00
tons per inch immersion at L.W.L,	12.36
Area immersed, midship section, square feet,	377.46
L.W.L. plane, square feet,	5,200.00
Coefficient of fineness, block,	.533
midship section,	.831
L.W.L.,	.808

GENERAL DESCRIPTION OF HULL.

The vessel is of the flush-deck type. There are two masts, fitted with wireless, signal yards, etc., and one smoke pipe.

Main Deck.—The main deck is continuous from stem to stern. On it are located the windlass, torpedo tubes, pilot house, deck house, towing engine and electrolyte tank. The deck house contains the flotilla commander's and commanding officer's quarters, the ship's and submarines' offices, galleys and sick bay. On top of the deck house there is an electric winch used for hoisting boats. There is also a steering platform and an acid tank, besides the boat stowage. On top of the pilot house and extending to the sides of the vessel is the bridge, fitted with steering wheel, searchlight and other navigational instruments.

Second Deck.—This deck extends from the bow to the engine space forward, and aft from the engine space to the stern. Foward are located the officers' storerooms and quarters, wardroom and machine shop. Aft are located the petty officers' and crew's quarters.

Platform Deck.—This deck extends from the bow to the boiler room, and from the engine space to the stern. Forward are located the general storeroom, submarine storerooms, torpedo stowage, refrigerator rooms, armory, and trimming tanks. Aft are crew's quarters.

Hold.—In the hold, forward and aft, are located trimming tanks, fuel-oil tanks, fresh-water tanks, magazines and stores. The chain lockers are forward, and the machinery spaces occupy the entire midportion of the hold. The engineers' storerooms are located in the shaft alley.

Battery.—The battery consists of a single twin-deck torpedo tube.

Boats.—

2 35-foot motor boats.
2 33-foot motor sailing launches.
2 21-foot motor dories.
3 17-foot pulling dories.
2 28-foot whale boats.
1 12-foot punt.

Complement.—

1 Division commander (submarine).
1 Commanding officer (tender).
14 Wardroom officers (submarine and tender).
174 Petty officers and crew (submarine and tender).
190 Total for one division.

Fire Main.—The fire main, size 2.5 inches, is supplied by one fire and bilge pump. Branches for the fire plugs are taken off at convenient locations. The eight fire plugs are located as follows:

No. 1, forecastle, main deck.
No. 2, starboard side amidships, main deck.
No. 3, port side amidships, main deck.
No. 4, aft of deck house, main deck.
No. 5, wardroom country, second deck.
No. 6, compartment C-6, near forge, second deck.
No. 7, compartment D-9, crew space, second deck.
No. 8, compartment D-5, crew space, platform deck.

Fresh Water.—Fresh water for drinking and washing purposes is stowed in tanks in the hold in compartment A-5, capacity 24 tons. Reserve feed water to the extent of thirteen tons is carried in addition to the supply for the ship's use.

Drainage System.—A 4-inch drain extends the entire length of the vessel with connections to the fire and bilge pump, also to all compartments and trimming tanks. Macomb strainers are fitted near the pump suction.

Ventilation.—Forced ventilation is used for the living spaces. There are two circuits, one forward and one aft. The forward blower is located in compartment B-5, the after blower in compartment D-9.

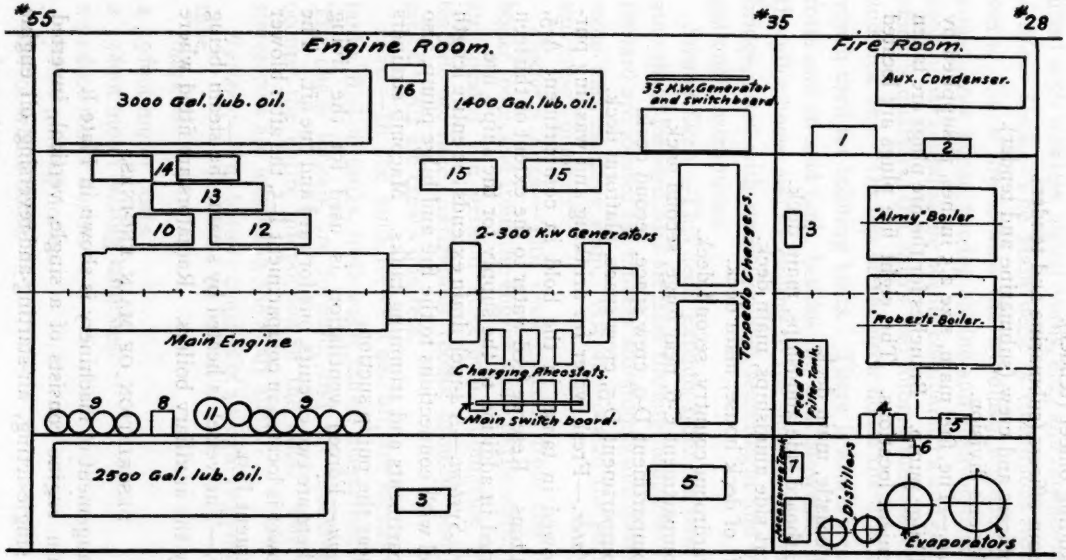
Heating.—The vessel is heated by steam, the steam being supplied by the auxiliary boilers. Radiators are fitted where necessary.

DESCRIPTION OF MAIN ENGINES.

The arrangement of machinery is shown in Plate I.

The main engine consists of a single, vertical, inverted, two-cycle, single-acting, air-starting-and-reversing oil engine

Machinery Diagram.



1 Distiller circ. pump.
2 Sanitary pump.
3 Fuel oil feed pump.
4 Main & Aux. feed pump.
5 Fire and bilge pump.
6 Evaporator feed pump.
7 Dist. fresh water pump
8 Westinghouse Compressor.
9 Engine air flask.
10 Air reservoir.
11 Water Cooler.
12 Oil Cooler.
13 Fresh water Tank.
14 Air accumulators.
15 Circ. W. & oil pump.
16 Lub. oil pump.

PLATE I.

of the Diesel type, with six working cylinders and two air compressors. The shaft horsepower is about 1,000 at 260 r.p.m. The six working cylinders and the air-compressor cylinders are are on the same fore-and-aft line, the air compressors being at the forward end of the engine, where the control station is located. The starting, stopping, reversing and changing of speed are all controlled by a hand wheel at this station. The pistons are of the stepped type, the upper piston being the working piston, the lower piston being the scavenger piston. From the scavenging cylinders the scavenging air at low pressure is carried to an external receiver whence it goes to the working cylinders, and overboard through the exhaust. The crank pit is entirely enclosed. Each working cylinder is provided with two scavenger valves, an air-starting valve, a spray valve and a relief valve. The spray valve atomizes and injects the fuel oil into the cylinder by means of high-pressure air supplied by the engine air compressors. All valves are actuated from a cam shaft located on top of the cylinders, which is operated from the main shaft through a vertical shaft and spiral gears. A pneumatic cylinder automatically places the cam shaft in the proper relation to the valves when the engine is started or reversed. Salt water is used to cool the cylinders of the engine, while the pistons are cooled by fresh water. The water for the pistons is carried through pipes with swinging joints. A small lubricating-oil pump furnishes the oil for the forced lubrication. The main-engine auxiliaries are duplicated, and are independently driven by electric motors.

The two main-engine air compressors are of the two-stage type, and supply high-pressure air for the fuel injection and for replenishing the air in the starting-air flasks. The ship's high-pressure system is connected for use in case of an emergency. The pump sets are independent units, each consisting of a centrifugal salt-water pump, a rotary fresh-water pump, a rotary lubricating-oil pump and a rotary fuel-oil pump. Each set is adequate for full power, the set not in use being a "stand by" set.

PLATE I.

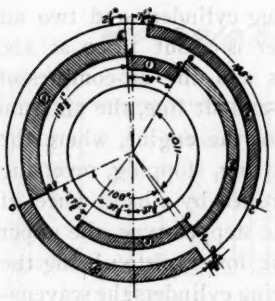

Exhaust Opening (Linear inches) 4½x12
" " Area in sq.ins. 54
① Spray air valve opens and closes. 35° total.
② Expansion of gases begins and is completed. 110° total.
③ Exhaust ports open & close - 75° total.
④ Scavenger valves " " " 100° "
⑤ Compression begins & ends. 112° "
⑥ Air starting valve opens & closes. 105° total.

PLATE II.

Plate II shows the valve diagram of the engines.

Main Engine Data.

Type	Nurnberg, reversible, 2-cycle, low-speed, Diesel.
Diameter working cylinders, inches	14.25
scavenging cylinders, inches	22.50
Stroke, inches	23.50
Diameter connecting rods, inches	5.25
Length connecting rods between centers, inches	47.00
Ratio of crank to connecting rod	1 to 4
Diameter of wrist pin, inches	7.00
Length of wrist-pin bearings, inches	7.375
Diameter of main journal, inches	9.00
Unit bearing pressure, with compression at 450 pounds per square inch and combustion pressure at 580 pounds per square inch, wrist pin, pounds	1,409.
crank pin, pounds	1,032.
Volume of working cylinder, cubic inches	3,750.
scavenger cylinder, cubic inches	9,350.
Clearance in working cylinder, cubic inches	314.
scavenge cylinder, cubic inches	1,290.
volume per cent., working cylinder	8.4
scavenge cylinder,	13.8
Linear clearance, working cylinder, inches	2.625
scavenge cylinder, inches	.1885
Cylinder constants, 1, 2, 3, 4, 5 and 6	0.009464
System of cooling cylinders	Salt water.
air compressors	Salt water.
pistons	Fresh water.

Lubrication.

Working cylinders	Forced feed, Detroit oilers.
Scavenger cylinders	Forced feed, Detroit oilers.
Air compressors, both stages	Drip feed.
Bearings, main and crank	Forced lubrication, pump set.

Air Compressor Data.

Type	Two-stage, stepped piston.
Number on engine	2.
Diameter first-stage piston, inches	11.875
second-stage piston, inches	3.875
Stroke, inches	17.75
Cranks are spaced 180 degrees apart, coincident with No. 5 and No. 2 engine cranks, respectively.	
Diameter of crank pin, inches	6.75
Length of crank-pin bearing, inches	7.0625
Diameter of connecting rod, inches	3.875
Length of connecting rod, inches	53.
Ratio of crank to connecting rod	1 to 5.97
Unit bearing pressure, pounds, at 1,000 pounds compression, wrist pin	548.
Unit bearing pressure, pounds, at 1,000 pounds compression, crank pin	247.

SHAFTING.

The shafting is in four sections, consisting of a line shaft in two sections and a crank shaft in two sections. The after section of the line shafting is supported by two bearings, the forward section by one bearing. The bearings are lined with white metal.

Shafting and Bearing Data.

Shaft.	Material.	Length, ft. and ins.	Diameter, inches.	Diam. holes, inches.
Crankshaft, 2 sections	Carb. steel	22–05.5625	9.0	3.25
Crank pins, main, engine	Carb. steel	0–10.25	9.0	3.25
Crank pins, air compressor	Carb. steel	0–07.25	6.75	2.25
Thrust shaft	Steel	12–10.125	8.0	Solid
Line shaft, 2 sections	Steel	33–00	8.0	Solid
Propeller shafting	Steel	16–10.5	8.0	Solid

Number of crankshaft bearings	10.
Length of main bearings, inches:	
Air compressor, No. 1 and No. 2	7.5625
Main engine, Nos. 3, 4, 5, 6, 7, 8 and 9	10.
Main engine, No. 10	7.3125

Length of thrust-shaft bearings, inches	8.
Collars on thrust shaft:	
Number	6.
Thickness, inches	1.625
Space between, inches	3.5
Outside diameter, inches	14.
Inside diameter, inches	8.
Bearing surface, square inches	622.
Thrust shoes, number	6.
effective thrust surface, square inches	486.
Line-shaft bearings, number 3, length, inches	10.
Stern-tube bearings—Material, wood:	
Length, forward, inches	17.
after, inches	32.

Propeller Data.

Type	True screw, solid.
Material	Manganese bronze.
Number of blades	3
Diameter, feet and inches	7–09.5
Pitch, feet and inches	6–00
Ratio of diameter to pitch	1.29 to 1.
Area, projected, square feet	14.25
helicoidal, square feet	15.90
disc, square feet	47.68
Height of lower blade above keel, inches	10.
Immersion of upper tip of blade, inches	55.

Condenser and Feed-Heater Data.

Main condenser, Worthington, one located in fireroom; 300 square feet cooling surface.

Feed heater, integral with Roberts boiler.

Evaporators, 36-inch Reilly multicoil, located in fireroom.

Distillers, 18-inch Reilly multicoil, one, No. 7, type D, located in fireroom.

Evaporator feed heater, Wainwright, horizontal, No. 1, type PP and SS.

Fuel-oil heater, one, size OA, single-tube, Schutte and Koerting, spirally corrugated film.

BOILERS.

There are two boilers, one a "Roberts," the other an "Almy" boiler. The original installation contemplated but

one boiler, but experience in service showed the necessity for two. The "Almy" boiler has been installed. The single smoke pipe will care for both boilers. The boilers burn oil with natural draft, are located alongside of each other, and are supplied by the same auxiliaries. Both boilers are water-tube boilers. The top of the smoke pipe is 49 feet 6 inches above the burners.

GENERATORS AND AIR COMPRESSORS.

The generators for charging the submarine batteries are two in number and are driven from the forward end of the main-engine crankshaft. A turbo generator set is used for the ship's lighting and power. There are also two air compressors for charging the air banks of the submarines, and the vessel's air banks, as well as the torpedoes, both electrically driven. Air for pneumatic tools is supplied by a low-pressure compressor.

STEAM AUXILIARIES.

Main air and circulating pump, one; combined, horizontal, simplex, "Worthington;" fireroom.

Main feed pump, one; vertical, simplex, "Blake;" fireroom.

Auxiliary feed pump, one; vertical, simplex, "Blake;" fireroom.

Main sanitary pumps, two; horizontal, duplex, "Blake;" fireroom.

Fuel-oil filling pumps, two; horizontal, duplex, "Blake;" engine room.

Fuel-oil feed pumps, two; horizontal, duplex, "Blake;" fireroom.

Lubricating-oil filling pumps, two; horizontal, duplex, "Blake;" engine room.

Low-pressure air-compressor pump, one; "Westinghouse," vertical; engine room.

Evaporating and distilling-plant pumps:
Distiller fresh-water, horizontal duplex; fireroom.
Evaporator feed, horizontal, duplex; fireroom.
Distiller circulating, horizontal, duplex; fireroom.
Steering engine, one; automatic worm-gear type, American Engineering Company, Phila., Pa.; second deck.
Anchor engine, one; Providence spur-gear type, American Engineering Company, Phila., Pa.; main deck, forward.
Towing engine, one; Automatic Steam Towing Machine, American Engineering Company, Phila., Pa.; main deck, aft.

ELECTRICAL AUXILIARIES.

Main air compressors, two; 90 H.P., open, General Electric Co.; engine room.
Main circulating pumps, two; 7 H.P., enclosed, Electro Dynamic Co.; engine room.
Main generators, two; 300 k.w., open, Crocker and Wheeler; engine room.
Forge blowers, one; 0.125 H.P., enclosed, Emerson Electrical Manufacturing Co.; workshop.
Machine-shop motor, one; 20 H.P., enclosed, General Electric Co.; workshop.
Fire and bilge pumps, one; 25 H.P., enclosed, and ventilated, General Electric Co.; engine room.
Ice machine, one; 5 H.P., open, General Electric Co.; fireroom.
Furnace and oil pump, one; 1.25 H.P., open, Westinghouse Electric and Manufacturing Co.; workshop.
Deck winch, one; 20 H.P., enclosed, General Electric Co.; upper deck.
Ventilating blowers, two; 1.125 H.P., semi-enclosed, B. F. Sturtevant; second deck.
Auxiliary generator, one; 35 k.w., open, turbo, General Electric Co.; engine room.

SHOP EQUIPMENT.

Machine Shop.—

1 screw-cutting back-geared extension-gap lathe, 7 feet between centers when extended, swing 32 inches.
1 tool-room lathe, 14.5-inch swing, 21 inches between centers.
1 screw-cutting back-geared tool-room lathe, 14.5-inch swing, 57 inches between centers.
1 8-inch back-geared precision lathe, 22 inches between centers, with milling attachment and grinders.
1 16-inch high-speed sensitive drill fitted for No. 2 Morse taper.
1 upright drill, up to 1.5 inches to 14 inches from edge of work, 9-inch traverse of spindle, fitted for No. 4 Morse taper.
1 column tool-room shaper, 15-inch stroke, 20-inch traverse.
1 double emery grinder on column, wheels 12-inch diameter, 2-inch face.
1 universal milling machine, with overhanging arm, 20-inch longitudinal feed, 17-inch vertical movement, 7-inch traverse, spindle fitted for No. 10 Browne and Sharp taper.
1 power hack saw, to take up 8-inch solids.

Woodworking Tools.—

1 pattern-maker's gap lathe, 36-inch swing through gap, 4 feet between centers when extended.
1 band saw, 32-inch.
1 hand-operated universal wood trimmer, on column.

Foundry and Blacksmith Shop.—

1 marine combination furnace, oil-burning, to melt brass in a crucible, to serve as forge for blacksmith and coppersmith.
1 blacksmith forge, 42 inches square, 24 inches high, with electric-driven air blast and tuyere blast gate.

TRIALS.

The contract required the following trials:

(*a*) A progressive trial over a measured-mile course at Provincetown, Mass., for standardizing the screw, extending from maximum speed, at least 12.25 knots, down to a speed of 8 knots.

(*b*) A full-speed trial of eight hours duration in the open sea, in deep water, at the highest speed attainable, the average for the eight hours to be not less than 12.25 knots.

(*c*) A fuel-consumption trial of four hours in the open sea, in deep water, at an average uniform speed of 11 knots, as nearly as possible, all auxiliaries in operation, the consumption to be such that the vessel will have a cruising radius of 3,400

TABLE I — U.S.S. FULTON STANDARDIZATION TRIAL DATA OCTOBER 31, 1914.

NO. OF RUN	TIME ON COURSE		SPEED IN KNOTS	R.P.M.	I.H.P.
	MINS.	SECS.			
5	8	49	6.81	141.3	766
6	7	43.3	7.77	146.6	840
7	8	9.6	7.35	148.3	688
MEAN OF GROUP			7.425	145.7	784
8	6	4.8	9.87	188.9	556
9	6	19.8	8.48	186.0	573
10	6	13.4	9.64	185.9	560
MEAN OF GROUP			9.618	186.7	566
11	5	26.4	11.03	221.5	694
12	5	21	11.21	224.4	758
13	5	17.9	11.32	224.5	—
MEAN OF GROUP			11.193	223.7	726
14	5	1.7	11.93	240.7	982
15	4	58.8	12.05	241.9	1012
16	5	2.4	11.90	240.7	912
MEAN OF GROUP			11.983	241.3	980
17	4	43	12.72	258.0	—
18	4	42	12.77	260.8	1098
19	4	40.6	12.82	259.2	1042
20	4	39.7	12.87	260.8	1044
21	4	44.9	12.64	257.6	994
MEAN OF GROUP			12.785	259.7	1051

Data on Steaming Trials—Table II.

Average results.	3.5-hour 11 knots.	8-hour, full speed.
Speed, knots	11.092	12.342
R.P.M., number	221.41	249.5
I.H.P.	831.	1,097.
Fuel-oil consumption per hour, main engine only, pounds	350.5	483.3
Fuel-oil consumption per hour, all machinery, lbs.	622.5	652.2
Fuel-oil consumption per knot at speed indicated, pounds	56.12	39.16
Fuel-oil consumption per knot guaranteed, pounds	73.5	41.5
Radius of action on fuel oil at indicated speed, knots	5,372.5	5,706.
Lubricating oil consumed, gallons	10.	13.
Radius of action on lubricating oil at indicated speed, knots	10,404.	20,351.
Turbo generators—Ampères	120.	53.4
Volts	120.	120.
R.p.m.	3,500.	3,587.5
Pressures—Boilers, pounds	154.	149.5
Feed, pounds	164.4	154.1
Oil to burners, pounds	136.4	144.3
Sanitary system, pounds	24.3	26.9
Vacuum in condenser, inches	16.4	17.4
Temperatures, degrees F.—Engine room	61.1	67.4
Fireroom	71.1	74.8
Outside air	55.	62.1
Feed tank	not taken.	123.3
Barometer, inches mercury	30.04	30.00
Double strokes or r.p.m.—After feed pump	18.	10.4
Fuel-oil pump	8.	8.5
Sanitary pump	50.3	50.6
Air and circ. pumps	28.3	22.00
Distiller circ. pump	60.6	.....
Ice machine	120.	.....
Pressures, lbs. per sq. in.—Evaporator steam	25.	.....
shell	3.	.....
Spray bottle	1,000.	1,076.3
Starting bottle	828.5	845.6
1st stage air comp., for'd.	76.4	43.1
aft	54.3	56.1
Scavenger receiver	8.	9.1
Low-pressure fuel pumps	1.4	1.9
Circulating water	8.9	11.3
Lubricating oil	24.9	24.7
Fresh water	10.	12.1
Temperatures, deg. F.—Scavenger receiver	122.	136.5
Circ. water to engine	58.	56.6
from engine	89.	94.3
Fresh water to engine	76.7	76.1
cooler	104.	113.4
Circ. water from FW., cool.	88.	91.8
Lubricating oil to engine	71.	74.9
cooler	82.	87.6
Circ. water from oil cooler	60.	59.9
R.p.m. of independent pump aft	1,507.	1,466.3
Main generators.—Volts	120.	121.1
Ampères	57.	50.
Water evaporated during trial, gallons	375.	.....

nautical miles, exclusive of the fuel oil carried for the submarines.

(*d*) Trials to determine the ability of the vessel to back satisfactorily, to test the steering gear, and to test the anchor gear. Reversing tests, the time to reverse not to exceed 15 seconds.

Trials were held October 31, and November 1, 1914, the data for which is given in Tables I and II. From the data in Table I the standardization curve, Plate III, was plotted.

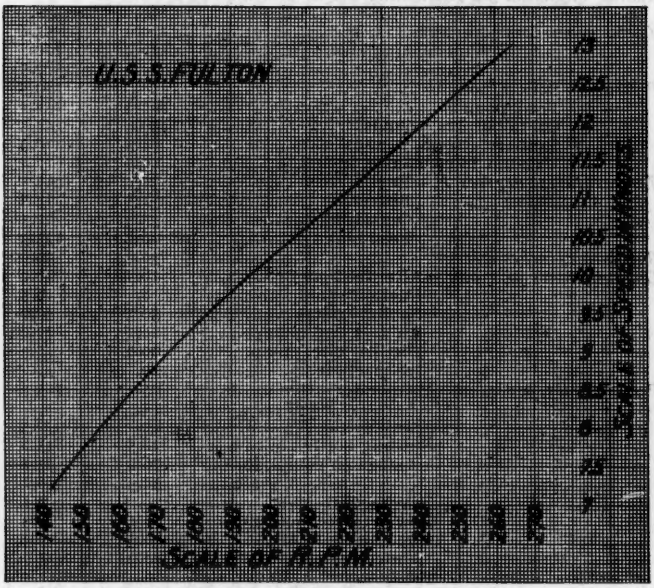


PLATE III.

A DEVELOPMENT OF A HIGH-GRADE ALLOY STEEL AT LOW COST.

By Lieutenant J. B. Rhodes, U. S. Navy, Member.

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1. It is the purpose of this paper to describe the qualities of an alloy steel which has been found to furnish high-grade castings and forgings from the same mix and at comparatively low cost. The principles involved are compatatively well known and no originality is claimed, but it is believed that the alloy described (and its complement) is being regularly manufactured for use for the first time.

2. In order to increase the output of a small open-hearth furnace it was desired to manufacture ingots for forging purposes without abnormally increasing the cost of castings. These ingots were to be used for forgings which are required to show minimum physical values in a transverse direction, as follows:

Tensile strength, 95,000 pounds per square inch.
Proportional limit, 65,000 pounds per square inch.
Elongation, 18 per cent.
Reduction of area, 30 per cent.

It was believed that the steel as cast would, if capable of showing the above values in forgings, give in properly annealed castings the following physical values:

Tensile strength, 85,000 pounds per square inch.
Yield point, 53,000 pounds per square inch.
Elongation, 22 per cent.
Reduction of area, 35 per cent.
Bend, 120 degrees around 1-inch diameter.

3. In going over the ground of previous experimenters the following points were determined:

(*a*) In order to obtain a steel of high physical properties with considerable ductility and machinability it is necessary to produce a pearlitic steel. High-grade steels are amorphous to a great degree and are fine grained. The microscope should show a uniform mixture of constituents. Although it is generally considered that true pearlitic is a eutectic of cementite and ferrite, it is known that a pearlitic structure is obtained in alloy steels in which the amount of pearlite is in excess of that due to carbon alone.

(*b*) Knowing that brittle ranges are found where carbon, nickel, chromium, manganese, etc., are present in the percentages necessary to give true eutectics, and knowing the superiority of ternary and quarternary steels, *i.e.*, steels containing chromium and nickel and those containing chromium, nickel and vanadium in addition to the carbon, it was decided to limit the percentage of each hardening or toughening element and to increase the pearlite by the use of additional elements. The following were considered:

(*a*) Carbon, which increases both strength and hardness, but which should not be present in larger amount than 0.60 per cent.

(*b*) Manganese, which increases the strength and hardness and acts as a deoxidizer, but which should not be present in larger amounts than 2 per cent.

(*c*) Nickel, which increases strength and toughness, but which should not be present in larger amounts than 5 per cent.

(*d*) Copper, which has an effect similar to that of nickel, but should not exceed 4 per cent.

(*e*) Chromium, which hardens the metal, increases its susceptibility to heat treatment, but which should not exceed 3 per cent.

4. Consideration of the above shows the possibility of obtaining steel of the grade desired by adding copper and manganese to the usual nickel-steel alloy. Steel containing more of the hardening agents than shown above are liable to

be brittle and treacherous; in fact, the amounts given above are considered rather high. Of these alloying agents carbon and manganese are the most easily obtained and the least costly. It was decided to limit the carbon to 0.35 per cent. and the manganese to 1.20 per cent., or about 0.60 per cent. higher than normal. It was found that nickel could be obtained in turnings from 3 per cent. nickel-steel in sufficient quantity to give 1 per cent. to 1.5 per cent. in the steel, and nickel with copper could be obtained in the form of monel-metal scrap and turnings containing approximately 65 per cent. nickel and 30 per cent. copper.

5. The increase in manganese by 0.6 per cent. or 6 pounds per 1,000 can be made by adding not more than 10 pounds per 1,000 of manganese, which in the form of 80 per cent. ferro-manganese, costs about $0.06 per pound for manganese, an increase of 60 cents per 1,000 pounds, or $0.006 per pound. Nickel-steel turnings are worth about $10.00 a ton more than ordinary scrap (as based on sale of own scrap), which means that nickel in this form is obtained for $0.05 per pound instead of $0.35 or $0.40 per pound. To obtain 1.0 per cent. to 1.5 per cent. nickel 15 pounds per ton are added, at a cost of 75 cents per 1,000 pounds, or $0.00075 per pound. To obtain .50–.75 copper, monel-metal scrap at $0.12 per pound is added. Two per cent. of monel metal gives .70 copper and 1.20 nickel, at a cost of $2.40 per 1,000 pounds, or $0.0024 per pound. The cost of these additions are as follows:

Manganese	$0.0006
Nickel	0.00075
Monel	0.0024
Total	$0.00375

So that for an increase of less than $0.004 per pound we obtain a steel equal in properties to a 3 per cent. nickel-steel. The composition of the steel may be taken to be as follows:

Carbon, . . .30 per cent. to .35 per cent.,
Silicon, . . .25 per cent. to .35 per cent.,
Phoshorus and sulphur, . Not over .05 per cent.,
Manganese, . . 1.00 to 1.20 per cent.,
Nickel, . . . 1.50 to 1.80 per cent.,
Copper, . . .50 per cent. to .80 per cent.,

which steel will show properties equal to those specified.

6. It is believed that the excess manganese prevents red shortness due to copper oxide by combining with any oxygen present in the bath. Castings are remarkably free from checks, cracks, blowholes and shrinks, and ingots are normal. The addition of chromium in sufficient quantity to give 0.50 per cent. chromium in the alloy increases the physical properties to those of ordinary chrome-nickel steel containing 1 per cent. chromium and 3 per cent. nickel.

7. The following table shows some physical tests and analyses of castings and forgings:

TEST BARS FROM COUPONS CAST ON BODY OF CASTINGS.

Heat No.	C.	S.	Mn.	Si	P.	Ni.	Cu.	Tensile.	Y. point.	Per cent. elong.	Per cent. red.	Bend deg.
201	.36	.036	.81	.31	.034	1.60	.76	92,283	55,767	22	37	120
204	.30	.037	.93	.36	.04	1.44	.85	92,334	55,003	21	31	120
207	.33	.030	1.02	.38	.035	1.15	.49	88,616	54,494	25	40	120
208	.34	.034	1.04	.36	.033	1.14	.52	92,691	56,277	22	34	120
214	.35	.035	1.11	.33	.042	1.34	.60	95,543	59,078	22	37	120
215	.30	.032	.93	.35	.043	1.12	.77	90,042	57,550	23	37	120
216	.27	.04	1.18	.35	.046	1.60	.61	92,691	55,003	22	36	120
218	.34	.042	1.15	.35	.035	1.58	.69	94,830	54,748	22	36	120
219	.27	.035	1.02	.30	.047	1.47	.42	85,510	56,022	23	34	120
223	.38	.029	1.02	.35	.049	1.10	.80	88,718	55,512	23	38	120
228	.30	.031	1.15	.38	.046	1.60	.56	96,510	59,587	23	37	120
230	.36	.036	1.18	.31	.049	1.37	.80	94,117	50,929	23	36	120
232	.36	.036	1.03	.35	.039	1.47	.63	90,144	55,225	23	34	120
233	.36	.035	1.03	.40	.04	1.73	.21	96,052	59,087	25	41	120
234	.32	.036	1.21	.36	.049	1.24	.65	91,366	55,003	25	44	120
236	.31	.032	1.15	.34	.043	1.54	.70	81,343	60,096	25	41	120
241	.31	.033	.96	.39	.042	1.27	.64	87,343	53,457	23	36	120
245	.31	.032	1.15	.37	.037	1.80	.62	92,181	55,288	23	36	120
248	.33	.032	.79	.42	.048	1.69	.65	87,191	54,239	23	38	120

TEST BARS CUT FROM BODY OF FORGINGS—TRANSVERSE.

(*Steel of same heats shown above.*)

Heat No.	Tensile strength.	Limit of proportionality.	Per cent. elongation.	Per cent. reduction.
A–410–6	102,469	72,421	21.65	37.6
	98,853	64,272	23.80	47.3
A–410–9	99,719	63,254	21.85	42.2
	97,529	63,254	26.45	52.4
A–410–10	100,585	68,347	17.35	30.78
	97,529	63,254	26.45	52.4
A–408–1	110,516	77,412	21.6	45.2
A–408–2	113,164	80,570	18.9	34.08
A–408–3	110,923	77,514	23.5	51.3
4 × 1	117,952	85,561	22.3	51.0
4 × 2	119,836	89,890	20.2	49.6

TEST BARS CUT FROM BODY OF FORGINGS—LONGITUDINAL.

A–779–1	105,423	72,319	24.8	56.4
A–779–4	95,135	67,226	28.7	65.7

FORGINGS OF SAME COMPOSITION CARRYING ABOUT 0.50 PER CENT. CHROMIUM.

A–749	130,379	117,608	19.0	47.6 Long.
6 × 1–1	141,940	107,050	16.0	42.0 Trans.
	146,417	113,739	15.2	45.0 Trans.
6 × 1–2	156,329	117,429	15.5	42.5 Trans.
	147,695	108,021	15.1	42.2 Trans.
6 × 3	126,151	106,543	16.6	40.0 Trans.
6 × 4	130,328	106,441	17.7	43.0 Trans.
6 × 5	130,226	110,516	19.6	48.5 Trans.

DESCRIPTION AND TRIALS OF U. S. S. *SACRAMENTO.*

(*Gunboat No. 19*).

By W. F. Sicard, Associate.

The construction of the *Sacramento,* Gunboat No. 19, was authorized by an Act of Congress, approved March 4, 1911.

Plans and specifications for this vessel were issued to prospective bidders, May 1, 1912, and bids were opened at the Navy Department, at noon, July 1, 1912.

The contract, awarded to The William Cramp and Sons S. & E. B. Co., of Philadelphia, Pa., at a price of $492,500.00, called for a vessel of about 1,425 tons trial displacement. It contained the usual provision embodied in the act entitled "An act to increase the naval establishment, of August 3, 1886." The vessel was to be completed and delivered to the Government within 21 months of the date of contract and there were the usual penalties for under speed and over time.

The following trials were required by the contract:

(*a*) A progressive trial over a measured course at Delaware breakwater, for standardizing the screws, extending from maximum speed down to a speed of 8 knots, about 14 runs over the course being specified. On this trial the guaranteed speed for the five highest runs was 12½ knots.

(*b*) A full-speed four-hour trial, in deep water in the open sea, at the highest speed attainable, the speed developed upon this trial to be not less than an average of 12½ knots an hour.

(*c*) A coal-consumption trial of 10 hours in the open sea in deep water, at an average uniform speed of 10 knots.

U. S. S. "SACRAMENTO" (GUNBOAT NO. 19).

(*d*) Trials to determine the ability of the vessel to back satisfactorily at full speed and at cruising speed; trials for stopping, and the usual trials for the steering gear and the anchor engine.

The guaranteed coal consumption per knot run, including coal necessary for all auxiliaries in use on trial, was not to exceed 155 pounds on trial (*b*) and 127 pounds on trial (*c*).

HULL DATA.

The *Sacramento* is a single-screw gunboat, built of steel. She has flush main and berth decks, and has forward and after orlop decks. The principal dimensions and characteristics of the hull, are as follows:

Length between perpendiculars, feet	210–00
on load water line, feet	210–00
over all, feet and inches	226–02
Beam, extreme, feet and inches	40–10⅜
at load water line, feet and inches	40–10⅜
Ratio length to beam	5.13
Draught, forward, feet and inches	11–06
aft, feet and inches	11–06
mean, feet and inches	11–06
Displacement, load draught, tons	1,431.47
per inch at L.W.L.	12.94
Area, immersed midship section, square feet	437.0
L.W. plane, square feet	5,437.5
Wetted surface, square feet	9,180.
Coefficient of fineness, block	.507
midship section	.950
L.W.L.	.630

Masts.—There are two steel masts, each about 105 feet 5¾ inches above the water line, and located at frames Nos. 24½ and 51½. They are fitted with wireless, signal yards, etc. The foremast has a searchlight platform 51 feet 6¾ inches above water line and there is a military top, 43 feet 6¾ inches above waterline. Both masts are fitted with grab-irons, for climbing.

Navigating Bridge.—The navigating bridge is immediately

above the chart house and extends from frame 19 to frame 22. There is a ladder on the after end, starboard side, leading to the superstructure deck.

Superstructure Deck and Chart House.—This deck extends between frames Nos. 19 and 28 and contains the chart house, from frames Nos. 19 to 23. On each side of the superstructure deck is mounted a 3-pdr. saluting-gun. The foremast passes through this deck a little forward of frame No. 26.

Main Deck.—The main deck, on which is located the battery, is a weather deck throughout. There are two deckhouses, one forward, between frames Nos. 19 and 28, containing the conning tower, and captain's cabin, stateroom, bath and pantry, the other between frames Nos. 34 and 42, containing smokepipe and ventilator inclosures, bread room to port and radio room to starboard, while the officers' and crew's galley occupy the after part.

Berth Deck.—The berth deck extends from stem to stern. Beginning forward, there is the paint room, shower and bathroom for wardroom officers and on each side seven wardroom officers' staterooms, the wardroom country being located at center of ship, between the staterooms mentioned. Next to the above, abaft frame No. 19, is the wardroom mess room, extending to frame No. 23, this is followed by the wardroom pantry, outboard of which, on the starboard side, is the navigator's stateroom, the executive officers' stateroom and the captain's and executive officers' office, while in similar locations, on the port side, is the paymaster's stateroom, the senior engineer officer's stateroom and the pay office. The above enclosures end at frame No. 31, and from here to frame No. 45 is crew's space, broken only by the boiler hatch, at center of ship, and in wake of this, on the port side, the firemen's washroom, with ladder leading down to the fireroom. The engineroom hatch extends between frames Nos. 42 and 48, and, on the starboard side, between frames Nos. 47 and 64, is the engineer's office, chief petty officers' washroom, and ship's stores, and, on the port side, the operating room, sick bay

and sick-bay bath. The above subdivisions end at bulkhead No. 54, and forward of this bulkhead, at center of ship, is the armory. The remainder of berth deck, abaft frame No. 54, is crew's space, crew's water closets and urinals and the prison.

Orlop Deck.—The orlop deck extends from stem to stern, being broken only to accommodate the coal bunkers, the boiler room and the upper engine room. On the starboard side, outboard of the engine room is the engineer's workshop and, similarly located, on the port side, is space for 12 chief petty officers. The remainder of this deck, aft, is crew's space, except for the refrigerating room, extending from frame No. 60 to 63.

Hold.—In the hold, from forward, aft, is the forward trimming tank, chain locker, provisions, magazine, stores, coal bunkers, boiler compartment, engine compartment, shaft alley, ammunition rooms, stores and after trimming tanks. The boiler compartment extends between frames Nos. 34 and 41, and the engine compartment between frames Nos. 41 and 48.

Inner Bottom.—The reserve feed-water tanks are in the inner bottom, the full width of the boiler compartment, and extend between frames Nos. 34 and 37.

COMPLEMENT.

The ship's complement will be approximately as follows:

Commanding officer	1
Wardroom officers	9
Chief petty officers	12
Seamen's branch	73
Artificer branch	10
Artificer branch (engine-room force)	33
Special branch	7
Commissary branch	4
Messmen branch	11
Total	160

BATTERY.

The battery consists of the following:

One 4-inch rapid-fire gun	Main deck.
One 4-inch rapid-fire gun	Main deck.
One 4-inch rapid-fire gun	Main deck.
Two 3-pdr. saluting guns	Superstructure deck.
Two 1-pdr. guns	Main deck.

SMALL BOATS.

The following boats are carried, all on main deck except the whale boats, which are carried in davits:

Two 30-foot steamers.
Two 30-foot motor sailing launches.
Two 28-foot whale boats.
One 16-foot dinghy.
One 12-foot wherry.

Coaling Gear.—The coaling gear consists of one electric winch with two gypsy heads, capable of lifting 1,600 pounds at a speed of 125 feet per minute, and four coaling booms of 1,600 pounds capacity each.

The capacity of the coal bunkers is as follows:

B–2, Port side wing bunker	53 tons.
B–3, Starboard side wing bunker	56 tons.
A–18, Port, athwartship bunker	91 tons.
A–19, Starboard, athwartship bunker	91 tons.
A–16, Port, forward bunker	65 tons.
A–17, Starboard, forward bunker	65 tons.
Total	421 tons.

Fire Main.—The fire main is supplied by the fire and bilge pumps in the engine room and the fireroom. It extends, on the orlop deck, through the machinery spaces forward to frame No. 19 and aft to frame No. 59. There are 3½-inch

risers at frames Nos. 19, 35½, 42½ and 60, and, in the engine room, an emergency connection to supply circulating water to the distillers. At frame No. 19 the fire main rises to the berth deck with a 2½-inch fire plug at frame No. 19, a 3½-inch riser at frame No. 16, a 2-inch fire-main relief valve forward of this riser, and a connection to deck scupper pipe at frame No. 12½. The riser at frame No. 35½ extends to the berth deck with 2½-inch plugs at frame 31½ and a 2½-inch riser, to deck above, at frame 28.

HULL AUXILIARIES.

Steering Engine.—There is a steering engine, located in the after end of the engine room. It is a double engine with two 9-inch diameter cylinders and 6-inch stroke. It is manufactured by the American Engine Co., of Philadelphia, Pa.

Anchor Engine.—The anchor engine, located forward on the orlop deck, is also manufactured by the American Engine Co. It is a double engine with 6-inch diameter cylinders and 8-inch stroke.

Sanitary Pumps.—There are two sanitary pumps, located on the port side of the engine room, for the sanitary and flushing system. They are of the centrifugal, electric motor-driven type, and each is capable of operating the system. Each has a capacity of 250 gallons of water at 1,500 r.p.m. and is driven by a 7½-H.P. General Electric Co. motor. The suction is 4 inches and the discharge 3 inches in diameter.

Fresh-Water Pumps.—There are two fresh-water pumps, for ship's purposes. They are of the centrifugal type and electric driven. Each has a capacity of 70 gallons of water at 1,800 r.p.m. and is driven by a 3-H.P. General Electric Co. motor.

Main Drain.—The main drain, 7 inches diameter, runs from the boiler compartment in a single pipe, on the port side, to the after end of the engine room, where it is connected to the main circulating-pump suction pipe. There are 7-inch diam-

eter valves at the bilge wells in engine and boiler compartments, operated in place and from the berth deck. There is also a valve at the connection with the circulating-pump suction pipe, as a safeguard against flooding the main drain from the sea.

Secondary Drain.—The secondary drain extends in a single line from a manifold at frame No. 25 to the after trimming tank. It is 5 inches diameter to the after end of the engine room, abaft which it is 4 inches in diameter to the trimming tank. The manifold at frame No. 25 has a 3-inch diameter suction from forward trimming tank, four 3-inch suctions from holds and a 5-inch suction from coal bunker. In the fireroom there is a 5-inch bilge suction, two 5-inch suctions from coal bunkers and a 4-inch connection to the fireroom fire and bilge pump. In the engine room there are two 3-inch connections to the double bottom, a 5-inch bilge connection, and a 4-inch connection to the engine-room fire and bilge pump. From the after end of the engine room the system consists of a single pipe, 4 inches diameter, leading through the shaft alley to the after trimming tank, with 3-inch branches to the hold, after trimming tank and shaft alley. The shaft-alley connection is fitted with a Macomb strainer. At the after end of the engine room there is a 5-inch suction, through a Macomb strainer, from the bilge.

Heating System.—The ship's heating system is arranged in five circuits, as follows:

No. 1, Officers' quarters and heaters of ventilating system No. 4.

No. 2, Heaters of ventilating systems No. 2 and No. 5.

No. 3, Crew's quarters (radiators).

No. 4, Galley, pantry and plumbing heaters and fresh-water gravity tanks.

No. 5, Sterilizing apparatus, and plumbing heaters in operating room.

All pipes are of drawn brass, iron-pipe size. Coils are of 1-inch diameter pipe.

MAIN ENGINE.

There is one main engine, designed to develop 950 indicated horsepower when making 125 revolutions per minute. The engine is of the direct-acting, vertical, inverted, three-cylinder, triple-expansion type, the engine turning to starboard when going ahead. The order of cylinders, beginning forward, is high-pressure, intermediate-pressure and low-pressure. The cranks are at angles of 120 degrees.

Cylinders.—The cylinders, valve chests and cylinder and valve-chest covers are of cast iron. The cylinders are not steam jacketed. The high-pressure and intermediate-pressure valve-stem guides are cast with the lower valve-chest covers, and the low-pressure valve-stem guide is bolted to the under side of its valve chest. The intermediate-pressure and low-pressure balance cylinders are cast with the upper valve-chest covers. The valve-stem guides are fitted with composition bushings.

Framing.—Each cylinder is supported by a cast-iron, inverted Y-shaped box frame and a Class B forged-steel column, 4½ inches in diameter. Each box frame is secured to the bedplate and to its cylinder by 1⅛-inch diameter bolts, 16 and 6 respectively. The columns are secured at bottom and top by four bolts, 1⅝-inch diameter and 1½-inch diameter respectively. The steel columns are on the working side of the engine. The Y frames carry facings for securing the crosshead guides and the reversing-shaft brackets.

Crosshead Guides.—The crosshead guides are of the bar type, of cast iron and hollow, for the circulation of cooling water. At the upper end each is bolted to a facing on its cylinder by two 1¼-inch-diameter fitted bolts, and at the lower end to a facing on the box column by two 1¼-inch fitted bolts.

Bedplate.—The bedplate is of the box type, of cast iron, in one casting, and supported on keelson plates. Proper facings are provided for main bearings, columns, box frames, etc.

Main Bearings.—There are six crank-shaft bearings, one

pair to each crank, and each has two 2¼-inch diameter cap bolts. Each bearing is composed of a bottom brass, two distance pieces, and a cap. The cap is of cast steel, Class B, and the other parts of composition. The cap and brass are lined with white metal and provided with oil channels. The bottom brass is cored for the circulation of cooling water.

Pistons.—All pistons are of the conical type, of cast-steel Class B and are fitted with cast-iron followers, bull rings and packing rings. The pistons have two packing rings cut obliquely in one place, and clamped by fitting into a recess in the bull ring. No piston rings are fitted.

Piston Rods.—The piston rods are Class B forgings, solid, and are accurately ground and polished. The upper end is tapered to fit the piston which is secured by a collar, nut and split key. A dowel pin, between piston and rod, prevents turning. The lower end is also tapered and secured to the crosshead by a shoulder, nut and a split pin.

Crossheads.—The crossheads are Class B forgings, secured to taper on lower ends of piston rods. Each is secured to the flange provided on the crosshead slipper by four 1⅛-inch-diameter stud bolts. The crosshead pins have a 1-inch-diameter axial hole extending from outer face to body of crosshead, and the pins are flattened 3/16 of an inch, on each side, to assist in retaining the lubricating oil.

Crosshead Slippers.—Crosshead slippers are of manganese-bronze and are faced with white metal. Each slipper grasps three sides of its bar guide and is secured in place, on the fourth side, by means of two liners and a manganese-bronze cap, through the medium of four 1⅛-inch diameter bolts. The effective ahead surface of the slipper is 9 by 15½ inches and the backing surface is 9 by 14 inches.

Connecting Rods.—The connecting rods are solid, of forged-steel, Class B, forked at the top to span the crosshead and "T" headed at the bottom to receive the crank pin box. The crosshead boxes are composition with composition liners and the crank pin boxes are lined with white metal.

Valve Gear.—The valve gear is of the Stephenson type, with double-bar links, and all valves are worked direct. There is one single-ported valve for the high-pressure cylinder, one for the intermediate-pressure cylinder and a double-ported slide valve, working on a false valve seat, for the low-pressure cylinder. The high-pressure valve rings are solid; the intermediate-pressure valves are fitted with bull rings, each bearing two narrow rings, cut obliquely. The intermediate and low-pressure valves are each fitted with the ordinary type of balance piston. The back of the low-pressure valve is provided with packing bars of hard cast iron, working against a relief frame. There is no independent cut-off gear other than the adjustable blocks in the reversing arms.

Eccentrics and Eccentric Straps.—The eccentrics are of cast iron, 3⅜ inches wide, each in two parts and bolted together by two 1⅛-inch-diameter bolts. They are rabbeted on each side for the flanges of the eccentric straps. The eccentrics are keyed to the shaft. The eccentricity is 2½ inches. The eccentric straps are of composition, lined with white metal ⅜ inch thick, dovetailed and hammered into place. The straps are in halves, secured together by two 1⅛-inch-diameter bolts. The eccentric rods, of forged-steel, Class B, have forked ends for spanning the links and "T" heads for securing to the eccentric straps. Each is secured by two 1⅛-inch-diameter studs. The link ends of rods are fitted with caps Class B, forged-steel, liners and composition bushings, and are adjustable. The eccentric rods are 1¾ inches diameter at top and 2½ inches diameter at bottom. The suspension links are 1 inch diameter at ends and 1¼ inches diameter at middle, and are of forged-steel, Class B. They are fitted with Class B forged-steel caps, liners and composition bushings and are adjustable. The link bars and blocks are Class B forgings and the gibs are of composition H.

Reversing Gear.—The reversing gear is of the vertical, direct-acting type, and fitted with follow-up gear. It is bolted

to the high-pressure cylinder of the main engine and is connected, through a connecting rod, to the reversing-shaft arm, the reversing shaft being connected to the main links through the reversing arms and the suspension links. The reversing-engine cylinder is 6 inches diameter by 11 inches stroke. The gear is controlled by a floating lever, the primary motion being taken from a hand lever at the working platform, the secondary motion being taken from the reversing arm.

Reversing Shaft.—The reversing shaft is solid, of forged-steel, Class B, 3¾ inches diameter. It is supported by double cast-iron brackets, secured to the high-pressure and intermediate-pressure columns. The reversing lever and reversing arms are of cast steel, Class B, the reversing arms being slotted for the reception of the blocks and graduated, the blocks being adjustable by means of a handwheel and screw, for altering the cut-off.

Turning Gear.—Provision is made for turning the main engine by hand.

Lubrication.—Lubrication is accomplished through distributing-oil boxes, located on the main engine cylinders. There is a 3½ by 1¾ by 4-inch steam pump, in the engine room, for keeping the distributing oil tank supplied with oil. This pump draws from the storage tanks and discharges to the distributing tank. The distributing oil boxes have connections to distribute the oil to the various bearing surfaces and are plainly marked with the names of the parts to which they lead. Each connection has an adjustable valve, a sight feed with glass protecting tube, and a wick tube. There is an electric-light installation behind the sight feeds.

Water Service.—There is a 1¼-inch-diameter water-service main, attached to the main engine, having a 1-inch-diameter branch to the thrust bearing, a ¾-inch-diameter branch with a hose connection for each crank-shaft bearing, two ¾-inch-diameter pipes for each crank pin, and a ½-inch-diameter branch to each hollow brass in crank-shaft bearings.

Main Engine Data.

Working pressure at H.P. chest, pounds by gage	200
Revolutions per minute, designed	125
Indicated horsepower of main engine, designed	950
Diameter of H.P. cylinder, inches	16
I.P. cylinder, inches	26½
L.P. cylinder, inches	44
Stroke, inches	26
Ratio, I.P. to H.P.	2.74
L.P. to H.P.	7.56
L.P. to I.P.	2.75

Cylinder clearances:	Per cent. of volume. Top.	Bottom.	Linear, inches. Top.	Bottom.
High-pressure	18.2	16.7	5/16	3/8
Intermediate-pressure	15.7	14.0	3/8	½
Low-pressure	9.2	9.3	25/64	9/16

H.P. valve, piston, diameter, inches	8
I.P. valve, piston, diameter, inches	14½
L.P. valve, double ported slide, length, inches	30¾
width, inches	42
Valve stems, diameter at stuffing boxes, inches	2⅛
below stuffing boxes, inches	1⅞
through valves, inches	1½
Balance pistons, I.P. diameter, inches	3½
L.P. diameter, inches	8
Piston rods, diameter, inches	4
length, surface to surface, feet and inches	3–11¼
Cylinder walls, H.P., I.P., and L.P., thickness, inches	1¼
Valve-chest liners, H.P., thickness, inches	⅞
I.P., thickness, inches	1
Cylinder relief valves, one each end each cylinder, H.P. diameter, inches	1½
I.P. and L.P. diameter, inches	2
Connecting rod, length center to center, feet and inches	4–10½
diameter at upper end, inches	3⅝
at lower end, inches	4⅜
crosshead end bolts (4), diameter, inches	2
crank-pin end bolts (2), diameter, inches	2¾

SHAFTING AND BEARINGS.

There is one line of shafting consisting of crank shaft, thrust shaft, two lengths of line shaft and one propeller shaft. All shafting is of forged-steel, Class B. Coupling bolts are of forged-steel, Class A.

Valve Diagram Data.—Normal.

Name of vessel	*Sacramento.*
Total I.H.P.	950 at 125 r.p.m.—200 pounds steam pressure.
Type and number of engines	One vertical, triple-expansion.
Diameter of cylinders and stroke	16″×26½″×44″; stroke 26″; ratio 1 : 2.74 : 7.56.
Valve gear	Stephenson link, double bar, direct connected, crossed rods.
Connecting rod	4′–10½″ C. to C. 4½ cranks.

	H.P.		I.P.		L.P.	
Eccentricity, inches	2½		2½		2½	
Travel of valve, inches	5		5		5	
Kind and No. of valves. Diameter of piston	one 8″ P.V. Inside.		one 14½″ P.V. Outside.		D.P.S.V. Outside.	
Steam taken on	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Width of port and clear length, inches	1½ × 19.1328		1½ × 36.55		2′–2″ × 36	
Steam lap, inches	1 7/32	1 9/64	1 5/64	1	1 3/64	61/64
Exhaust lap, inch	0	+ 1/8	0	+ 15/32	0	+ 27/64
Angular advance, degrees	37½		35½		36½	
Steam lead, angular, degrees and minutes	8–45	10–30	10–10	11–45	12	14–15
linear, inch	5/16	3/8	3/8	29/64	29/64	17/32
Cut-off, in inches	19 3/8	17 3/8	20 7/16	18 9/16	20 5/16	18 3/8
	.706		.75		.748	
decimal of stroke	.745	.668	.786	.714	.781	.716
Release in decimal of stroke	.916	.937	.925	.942	.921	.935
Exhaust lead, in inches	1 33/64	1 1/32	1 29/64	63/64	1 31/64	1 5/64
Compression, in inches	3½	3⅜	2⅞	3½	3 1/16	3½
decimal of stroke	.125	.139	.110	.125	.118	.125
Steam opening, inches	1 9/32	1 33/64	1 27/64	1½	1 29/64	1 35/64
Exhaust opening, inches	1½	1½	1½	1½	2–2	2–2
	4,316		5,600		7,635	
Velocity steam, feet per minute; piston speed, 541.66 feet per minute	4,445	4,187	5,750	5,450	7,870	7,400
Velocity exhaust, feet per minute; piston speed, 541.66 feet per minute	3,800		4,670		5,725	
Velocity of steam through throttle	5,560 feet per minute.....5-inch diameter.					
exhaust to condenser	4,660 feet per minute.....15-inch diameter.					

Crank Shaft.—The crank shaft is of the built-up type, having the shafts and pins keyed and shrunk on the crank arms. There are axial holes through shaft and crank pins. The cranks are 120 degrees apart.

Shaft Data.

Crank shaft, total length, feet and inches	13–8¼
diameter, inches	8½
axial hole, diameter, inches	3½
coupling disc (1), diameter, inches	17
thickness, inches	2½
bolts, number in coupling	6
(taper), diameter at face of flange	$2\frac{3}{16}$
journals, diameter, inches	8½
length, inches	9¼
crank pins, diameter, inches	8½
length, inches	9⅜
axial hole, diameter, inches	3½
webs, width, inches	16½
thickness, H.P., inches	5¼
I.P., inches	6¼
L.P., inches	7
Thrust shaft, total length, feet and inches	8–0
diameter, inches	8½
diameter axial hole, inches	3½
collars, number	6
diameter, outside, inches	14
inside, inches	8½
thickness, inches	1¼
space between, inches	4
Line shaft, number	2
length each, feet and inches	19–0
diameter, inches	8
diameter axial hole, inches	3½
Propeller shaft, length, feet and inches	21–5½
diameter, inches	9
diameter axial hole, inches	3½
length, taper for propeller hub, inches	18¼
taper, inches diameter per foot length	1
axial hole through taper, diameter, inches	2
casing at bearings, thickness, inch	0⅝
brass sleeve through stern tube, thickness inch	0⅛
Couplings, diameter, inches	17
thickness, inches	2½
bolts to each	6
pitch circle, diameter, inches	13

exhaust to condenser.....5,500 feet per minute.....5-inch diameter.
.....4,660 feet per minute.....15-inch diameter.

Thrust Bearing.—The thrust bearing is of the usual horseshoe type, of cast iron, with a steady bearing at each end. It rests on a cast-iron sole plate, securely bolted to the foundations and is adjustable, in a fore-and-aft direction by means of wedges, fitted between the ends of the bearing and lips, cast on the sole plate. There are six cast-iron horseshoes, faced with white metal which is grooved for the proper distribution of oil, and the horseshoes are adjustable, by means of nuts on 1¾-inch-diameter side rods. The steady bearings have cast-iron bottom boxes and caps, both lined with white metal, and each cap is secured by two 1½-inch-diameter bolts. The body of the thrust bearing forms an oil well which is cooled by a water-service pipe, and there are stuffing boxes and glands, at the ends of the steady bearings, to retain the oil. Provision is made for circulating water through each horseshoe.

Spring Bearings.—There are three spring bearings of cast iron, lined with white metal. They are fitted, at the ends, with stuffing boxes packed with lamp wick.

Stern-Tube Bearings.—There is a forward and an after stern-tube bearing, connected by a cast-iron stern tube. Each end of stern tube is fitted with a composition sleeve, in which the lignum-vitae bearing proper is installed.

Bearing Data.

Main bearings, white metal lined, number	6
diameter, inches	8½
length, inches	9¼
Thrust bearing, white metal faced shoes, number	6
steady bearings, number	2
diameter, inches	$8\frac{1}{2}-\frac{8}{1000}$
length, inches	8
Line shaft (spring) bearings, white metal lined, number	3
diameter, inches	$8-\frac{1}{64}$
length, inches	16
Stern-tube bearings, lignum-vitae lined,	
forward bearing, diameter, inches	10¼
length, inches	15¾
after bearing, diameter, inches	10¼
length, inches	30½

PROPELLER.

There is one right-hand, four-bladed, manganese-bronze propeller with blades cast with hub. The blades are machined true to pitch and the hub has a taper fit on the shaft and is secured by one key and a nut. The taper of shaft is 1 inch in diameter per foot in length.

Propeller Data.

Diameter of propeller, feet and inches	9–08
hub, inches	22
Pitch, feet and inches	11–06
Ratio, diameter to pitch	00.84
Area, projected, square feet	29.24
helicoidal, square feet	36.32
disc, square feet	73.39
Ratio, projected to disc area	.398
helicoidal to disc area	.494
Distance, upper tip of blade from hull of ship, inches	16

CONDENSING APPARATUS.

Main Condenser.—There is one main condenser, cylindrical in form, located on the port side of the engine room. The circulating water passes through the tubes, entering at the bottom and passing out at the top. The tubes are straight, rolled into the tube sheet at the back and packed, with the usual glands, at the circulating water-entering end. The principal dimensions are as follows:

Inside diameter, feet and inches	3–06
Thickness of shell, steel, inches	00¼
Length between tube sheets, feet and inches	6–01
Thickness of tube sheets, inches	1 and 1¼
Tubes, number	1,113
diameter, outside, inches	00⅝
thickness, mils	65
Cooling surface, square feet	1,107
Main exhaust opening, diameter, inches	15
Auxiliary and dynamo-exhaust opening, inches	6
Air-pump suction, diameter, inches	4½
Circulation-water inlet and outlet, diameter, inches	8
Relief valve, diameter, inches	3½
Heads, cast iron, thickness, inches	0¾
Test pressure, shell, pounds per square inch	30

Main Air Pump.—The condenser is provided with a Blake vertical twin-bucket beam air pump with one steam cylinder 8 inches diameter, two water cylinders 16 inches diameter each and a common stroke of 12 inches. The suction and discharge pipes are 6 inches and 5 inches diameter respectively.

Main Circulating Pump and Engine.—There is one double-inlet, centrifugal circulating pump, operated by a vertical, single-cylinder engine. The crank shaft of the engine is forged-steel, Class A, and the impeller shaft is manganese bronze. The crank shaft is supported by two composition bearings lined with white metal and the impeller shaft by two bearings lined with lignum-vitae. The engine is enclosed, to just below the cylinder, to prevent the escape of oil from the forced-lubrication system. The forced-lubrication pump is operated by an eccentric on crank shaft, located between the flywheel and adjacent bearing. The main features of pump and engine, are as follows:

Circulating-pump engine, number of cylinders	1
cylinder, diameter, inches	5
stroke, inches	6
impeller, diameter, inches	28
width at tip inches	1¼
type, inclosed.	
Suction opening, diameter, inches	8
Discharge opening, diameter, inches	8

Feed and Filter Tank.—There is a feed and filter tank, of combined capacity of 450 gallons, located on the port side of the engine room. The tank is of Class C steel, rectangular in shape, 3 feet 6 inches long, 3 feet 5 inches wide, and 5 feet 1 inch high. The tank is thoroughly galvanized and covered with 1½-inch non-conducting material which is lagged with galvanized sheet iron, 14 mils thick. The connections are:

One main air-pump discharge, diameter, inches	5
Three vapor-pipe connections, diameter, inches	1½
One main feed-pump suction, diameter, inches	3
One auxiliary feed-pump suction, diameter, inches	3
One drain from heating system and whistle, diameter, inches	1¼

One drain from evaporator steam traps, diameter, inches	1
One feed-heater drain, diameter, inches	1½
One drain from scuttle-butt, diameter, inches	1
One overflow connection, diameter, inches	3

ENGINE-ROOM AUXILIARIES.

Auxiliary Condenser.—There is one auxiliary condenser, fitted with a turbo-circulating pump, and an independent air pump, located on the starboard side of the engine room. The condenser is cylindrical, with a steel shell, composition tube sheets and cast-iron water heads. The tubes are expanded into the tube sheets at the circulating-water end and are packed, with the ordinary type of gland, at the back. General particulars are as follows:

Shell, diameter inside, inches	24
thickness, inches	$0\frac{5}{16}$
Thickness tube sheet, front end, inches	1⅛
back end, inches	0⅞
Length between tube sheets, feet and inches	3–6½
Tubes, number	384
outside diameter, inches	0⅝
thickness, mils	65
Cooling surface, square feet	222
Circulating-water connections, diameter, inches	4
Exhaust connection, diameter, inches	7
Test pressure, pounds per square inch	30

Auxiliary Air Pump.—There is one auxiliary air pump; Blake, vertical, simplex, featherweight, 4½ by 8 by 6 inches, having a 3½-inch-diameter suction and a 3-inch-diameter discharge, for use with the auxiliary condenser.

Auxiliary Circulating Pump.—There is one auxiliary circulating pump. It is a Worthington, horizontal, volute pump with an impeller 8 inches in diameter. The impeller has 7 vanes. Diameter of suction, 5 inches; discharge, 3 inches. The pump and impeller are of composition "G."

STEAM, EXHAUST AND FEED PIPES.

Main Steam Pipes.—A 3½-inch-diameter main steam pipe leads from the stop valve on each boiler and, uniting in the

boiler compartment into a 5-inch-diameter pipe, passes through the after fireroom bulkhead, with a stop valve on the engine-room side of the bulkhead, and leads to the main engine throttle valve. There is no main steam separator.

Auxiliary Steam Pipe.—At each main boiler stop valve there is a 3-inch-diameter auxiliary steam stop valve from which a pipe leads aft, on each side of the boiler compartment, passes through the engine-room bulkhead, and supplies the auxiliaries on its side of the ship. In the boiler room the auxiliary steam pipes are cross-connected forward of the boilers, and from this cross-connection a 2-inch-diameter branch supplies steam to the forward auxiliaries.

Auxiliary Exhaust Pipes.—There is the usual auxiliary exhaust system with connections to the main and auxiliary condensers, feed-water heater, and the atmosphere. The dynamo turbines can also exhaust to the atmosphere.

Feed and Fresh-Water Systems.—The main feed pump, located in the engine room, draws from the feed tank, the condenser, and the reserve feed-water tanks. It discharges through, or by-passes, the feed-water heater and leads to the main feed-stop and check valves on the boilers. The auxiliary feed pump, in the boiler room, draws from the feed tank and the starboard and port reserve feed-water tanks, discharging to the auxiliary feed-stop and check valves on the boilers and to the reserve feed-water tanks. All feed-discharge pipes are fitted with air chambers at the boilers.

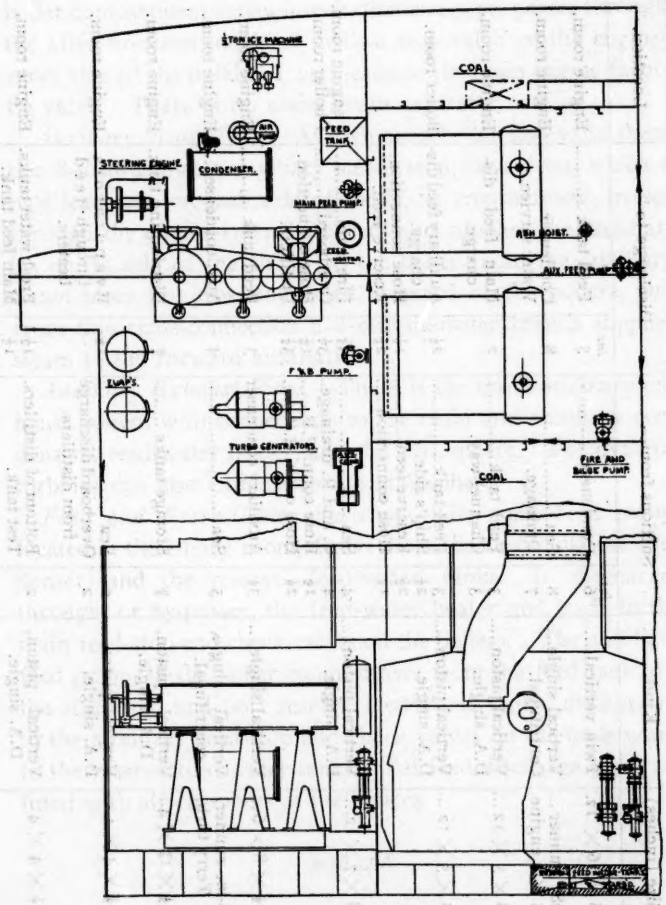
BOILERS.

There are two Babcock and Wilcox water-tube coal-burning boilers, in a common watertight compartment, arranged as shown. The boilers are designed to supply the necessary steam for operating all machinery, at full power, under natural draft. They are fitted with inswinging furnace and ash-pit doors.

The uptakes are of the usual design. There is one smoke-

U. S. S. Sacramento.—Pumps and Pump Connections.

No.	Pumps.	Size (inches).	Kind.	Suctions from—		Discharge to—		Location.
				Ins.		*Ins.*		
1	Main air.....	8 × 16 × 12..	Twin vertical.....	6	Condenser	5	Feed tank	Engine room.
1	Main circulating	28″ runner ... 5 × 6 engine.	Vertical single..... Centrifugal.	8 7	Sea..... Drainage.	8	Condenser ..... Water service.	Engine room.
1	Main feed	8 × 5 × 12 ...	Vertical piston..... D.A. single.	3 3	Feed tank..... Channel-way.	2	Main feed discharge.	Engine room.
1	Auxiliary feed ..	8 × 5 × 12 ...	Vertical piston..... D.A. single.	3 2½ 2½ 1½	Feed tank..... Ship's side. Reserve feed..... Hose connection	2½ 1½ 2½	Auxiliary feed discharge. Hose connection. Reserve feed.	Boiler room.
2	Fire and bilge....	8 × 7 × 12....	Vertical piston..... D.A. single.	4 4 1½ 2½	Sea..... C. and R. drainage.. Hose conn. F.F..... Hose conn. E.R.....	3½ 3½ 1½ 3½	Fire main..... Overboard..... Hose connection. Distillers.	1 engine room. 1 boiler room.
1	Auxiliary air. ...	4½ × 8 × 6 ...	Vertical single.....	3½	Auxiliary condenser	3	Feed tank	Engine room.
1	Auxiliary circulating.	8-in. runner, Terry turb,	Vertical single, centrifugal.	5	Sea	4	Aux. condenser ...	Engine room.
1	Lubricating oil..	3½ × 1½ × 4...	Vertical piston, D.A. single.	½	Storage tanks.....	½	Distributing tank.	Engine room.
1	Evap. feed.....	3½ × 4 × 4.....	Vertical piston.... D.A. single.	2 2	Sea ..... Overboard discharge from distillers.	1½	Evaporators through feed heaters.	Engine room.
1	Distiller fresh water.	3½ × 4 × 4 ...	Vertical piston, D.A. single.	2	Bottom of distillers. Test tank.	1½ 1½ 1½	Fresh water tanks. Main feed tank. Reserve feed tanks	Engine room.



U.S.S.SACRAMENTO.

pipe, 76 feet in height above the base line and oval in section, the outside length being 7 feet 5 inches, and the outside width 5 feet 8 inches.

Boiler Data.

Number of boilers	2
Working pressure, pounds per square inch, by gage	215
Test pressure, pounds per square inch, by gage	325
Height to top, external, feet and inches	10–11¼
center of drum, feet and inches	10–04¼
Length on floor, feet and inches	10–01
Width on floor, feet and inches	9–00½
Drum, inside diameter, inches	36
length, feet and inches	9–06¼
thickness, inch	00½
Number furnaces, each boiler	1
furnace doors, each boiler	2
Length of grate, feet and inches	7–00
Width of grate, feet	7.71
Grate surface, square feet, one boiler	54
total, square feet	108
Heating surface, one boiler, square feet	1,908
total, square feet	3,816
Ratio, heating surface to grate surface	35.33
Number of headers, each boiler	15
2-inch tubes, each boiler	350
4-inch tubes, each boiler	15
Exposed length of 2-inch tubes, feet	9
Area of smoke pipe, square feet	15.55
Ratio grate surface to smoke-pipe area	6.94
Diameter, main steam, boiler stop valve, inches	3½
auxiliary steam stop valve at boiler, inches	3
main and auxiliary feed stop and check valves, inches	2
surface blow valve, inches	1½
bottom blow valves (two), inches	1½
safety valve, duplex, inches	3

EVAPORATING AND DISTILLING APPARATUS.

The evaporating and distilling plant is located in the after end of the engine room. There are two vertical evaporators, arranged to operate in single or double effect; two distillers in the engine-room hatch; an evaporator feed-water heater; an evaporator feed-pump, distiller fresh-water pump, and all accessories. The plant has a combined normal capacity of 5,000 gallons of fresh water per twenty-four hours and, when clean, an overload capacity 40 per cent. in excess of the normal capacity. Circulating water for the distillers is supplied

by the engine-room fire and bilge pump, and there is an emergency connection from the fire main.

Evaporator Data (each).

Type and number	2 No. 12, Reilly, vertical, multi-coil Navy Type
Diameter, inside, inches	36
Length over all, feet and inches	6–06
Coils, number	12
Heating surface, square feet	73.80
Steam connection, diameter, inches	2
Vapor connection, diameter, inches	2½
Feed connection, diameter, inches	1
Blow-off connection, diameter, inches	2½

Distiller Data (each).

Type and number	2 Reilly, vertical, multi-coil, Navy Type.
Diameter, inside, inches	24
Length over all, feet	5–0
Coils, number	8
diameter, inches	1
thickness, inches	.065
Cooling surface, square feet	49.2
Circulating-water connections, diameter, inches	2½
Vapor inlet, diameter, inches	2½
Drain, diameter, inches	0¾

Evaporator Feed-Water Heater.—There is one cylindrical evaporator feed-water heater of the bent-tube type. The shell is of copper, tubes Admiralty metal, tube sheets rolled Naval brass, and head of composition G. The particulars are as follows:

Heating surface, square feet	14.24
Inside diameter of shell, inches	9.5
Thickness of shell, inch	0.125
Length extreme, feet and inches	3–8.625
Number of bent tubes	9
Tubes, outside diameter, inches	2
thickness, mils	72
Inlet and outlet, diameter, inches	1½
Vapor connections, diameter, inches	3
Tube sheet, diameter, inches	13¼
thickness, inch	0⅞
Test pressure, pounds per square inch	50

WORKSHOP.

The machine shop, suitably equipped for this class of vessel, is located on the orlop deck, off the engine room, on the starboard side, between frames Nos. 41 and 48. Machine tools, each driven by an independent, inclosed, variable-speed, electric motor, and all with the most modern attachments and necessary tools and appliances, are installed as follows:

No.	Description.	H.P. of motors.
1	16¾-inch by 32⅜-inch swing, extension-gap lathe, 4 feet between centers	2 H.P.
1	Column, tool-room shaper, 15-inch stroke by 15-inch traverse	1 H.P.
1	Upright drill, to drill up to 1½-inch diameter and to at least 10½ inches from edge of work, spindle fitted for No. 4 Morse taper, with reducers, sockets, etc.	2 H.P.
1	Double emery grinder, on column, with attachment for surface grinding.	
2	Machinists' swivel bottom vises.	
1	Steel, blacksmith forge, portable folding type.	

ELECTRIC PLANT.

The electric plant consists of two 25-kilowatt generating sets, located in the starboard side of the engine room. Each generating set consists of a direct-current General Electric generator, compound wound, furnishing a pressure of 125 volts. The generator is direct connected to a horizontal Curtis steam turbine of the single-stage condensing or non-condensing type, with three rows of buckets mounted on the same wheel. A row of intermediate or stationary buckets are assembled between each two rows of revolving buckets. The nozzles are contained in a separate casting and are not adjustable. Steam is admitted through a strainer in the emergency-valve chamber to the steam chest and is passed through double-balanced poppet valves, operated by the governor, direct to the steam nozzles, thence through the turbine to the exhaust. For condensing operation the turbines are, in addition, equipped with a hand-operating valve controlling steam to supplemental nozzles, which may be used to increase the

capacity of the machine when operating non-condensing, or with low pressure or vacuum. Supplemental nozzles are controlled by the governor. The turbines are designed to operate under a steam pressure of 200 pounds per square inch and 25 inches of vacuum when exhausting to the condenser. The sets operate, normally, at 4,500 r.p.m. Both main and outboard bearings are forced-feed lubricated, from a geared pump, driven from a worm and gear on the outboard end of the generator shaft. In emergency the outfit can be operated without forced lubrication. All lighting feeders are energized from the switchboard, located in the starboard corner of the engine room, at frame No. 46. In addition to supplying 300 fixtures installed for lighting the vessel, truck lights, running lights, night signalling set, one 18-inch hand-controlled searchlight and interior communication, current is furnished for wireless outfit, ventilating motors, potato peeler, fresh-water pump motor, sanitary-pump motor, workshop motors and deck-winch motor.

REFRIGERATING PLANT.

There is a ½-ton vertical, steam driven, Allen dense-air ice machine located in the engine room on the port side. The cooling pipes from the machine are led to the ice tank, scuttle butt and the cold-storage room. The ice machine is provided with a recooler. The cold-storage room, located on the berth deck between frames Nos. 60 and 63, on the port side, is subdivided into meat room, crew's refrigerating room and officers' refrigerating room. The capacities of these compartments are respectively, 293, 125 and 74 cubic feet, while the ratios of capacities to cooling surfaces are respectively 2.1, 2.25 and 2.1. The piping is arranged for the usual cut-outs, etc. A connection is provided for furnishing circulating water from the sanitary-pump discharge. There is a standard 4-can ice-making box, close to the ice machine. The scuttle butt is located on the berth deck between frames Nos. 51 and 52.

TRIALS.

The trials of the *Sacramento* were run over the course off the Delaware Breakwater, March 31 and April 1, 1914. The standardization trials were run first, and it was decided to run these trials as follows:

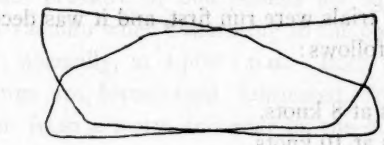
Three runs at 8 knots.
Three runs at 10 knots.
Three runs at 12 knots.
Four runs at highest speed attainable.

The standardization runs were begun at 9:39 A. M., March 31, 1914, and were completed at 1:30 P. M. the same day. The conditions were fair, with a gentle to moderate breeze from E. ½ S. with choppy sea from same direction, but not enough sea to cause the ship to roll. The results of the standardization runs were as follows:

No of runs.	Speed, knots.	R.P.M.	I.H.P.
1	6.923	74.14	195
2	8.511	76.74	216
3	6.693	75.11	213
4	10.946	97.44	427
5	8.918	98.34	449
6	11.029	99.51	445
7	11.465	121.00	838
8	12.645	122.88	860
9	11.830	121.15	857
10	13.230	133.24	1,129
11	13.260	134.27	1,164
12	12.574	129.67	994
13	13.148	130.49	1,027
14	12.380	130.71	1,031

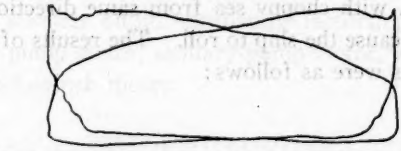
H.P.CYLINDER

M.E.P..... 95.5
M.R.P..... 14.77
I.H.P.....317.
Scale of spring..120



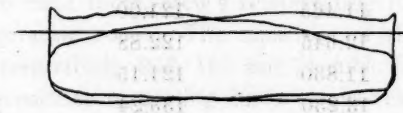
I.P.CYLINDER.

M.E.P..... 38.5
M.R.P..... 14.06
I.H.P.....363.
Scale of spring.. 40



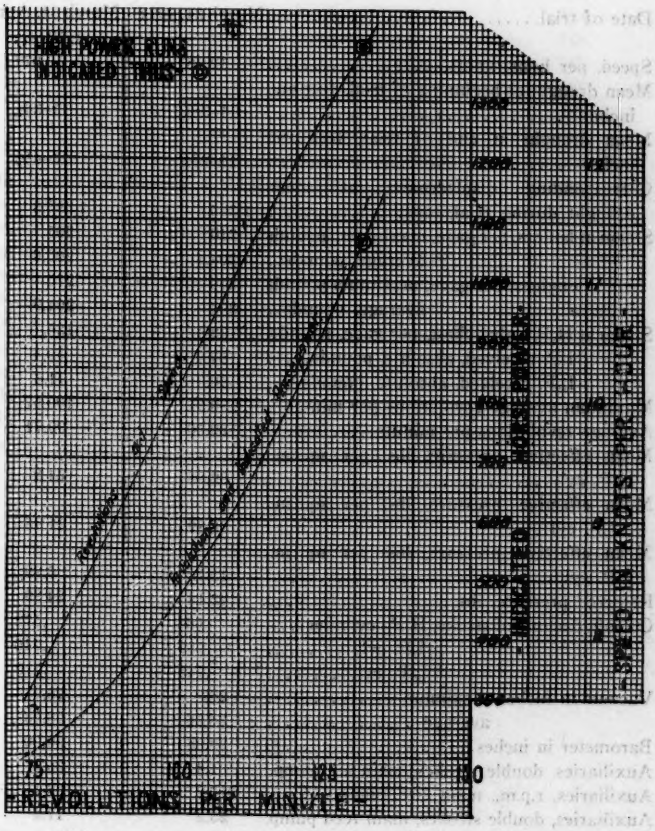
L.P.CYLINDER.

M.E.P..... 14.6
M.R.P..... 14.60
I.H.P.....376
Scale of spring.. 20



U.S.S.SACRAMENTO.
FOUR HOUR FULL POWER TRIAL

Revolutions per minute, 129.75
Total I.H.P..... 1056
" M.R.P..... 43.43
Card No.3. 3/31/14



U. S. S. "SACRAMENTO."—STANDARDIZATION TRIALS, MARCH 31, 1914.

Data of 4-Hour Full-Power Trial and 10-Hour 10-Knot Trial.

	4-hour full-power trial. (*b*)	10-hour 10-knot trial. (*c*)
Date of trial	March 31, 1914	March 31, April 1, 1914.
Speed, per hour, knots	12.781	10.077
Mean draught at beginning of trial, feet and inches	11–3⅝	11–3⅜
Mean draught at end of trial, feet and inches	11–2⅞	11–4½
Corresponding displacement at mean draught during trial, tons	1,395	1,392.3
Steam at boilers, per gage, maximum, pounds	210	205
average, pounds	204.4	200.6
engines, per gage, maximum pounds	210	210
average, pounds	204	201.9
Steam at H.P. valve chest, per gage, pounds.	192.3	126.
I.P. received, absolute, pounds	67.4	40.4
L.P. received, absolute, pounds	20.3	11.8
Maximum average revolutions per minute	130.61	100.57
Average revolutions per minute	129.50	99.71
Mean effective pressure, lbs. per sq. in., H.P. cyl	92.88	61.2
Mean effective pressure, lbs. per sq. in., I.P. cyl	38.89	21.95
Mean effective pressure, lbs. per sq. in., L.P. cyl	13.72	6.89
Referred pressure, lbs	42.14	24.39
Cut-off, decimal of stroke, H.P. cylinder	.664	.572
I.P. cyl	.750	.710
L.P. cyl	.748	.710
Vacuum in inches, maximum	28.8	28.9
average	28.56	28.23
Barometer in inches	29.98	30.00
Auxiliaries, double strokes, main air pump.	31.4	25.3
Auxiliaries, r.p.m., main circ. pump	210.2	166.1
Auxiliaries, double strokes, main feed pump.	23.2	17.2
Auxiliaries, r.p.m., sanitary pumps	1,508.	1,516.
dynamo steam pressure, pounds.	198.	197.9
exhaust	66.3	24.2
ampères	180.1	178.2
Temperatures, injection, degrees	38.	37.3
discharge	70.8	63.5
air pump discharge	67.5	59.9

	4-hour full-power trial. (*b*)	10-hour 10-knot trial. (*c*)
Temperatures, feed	195.2	211.8
outside air	42.	42.
engine-room, working level.	73.6	78.
fireroom, working level	61.3	79.9
I.H.P. high-pressure cylinder	307.	156.
intermediate-pressure cylinder	362.	159.
low-pressure cylinder	353.	136.
total	1,022.	451.
all machinery per square foot of grate surface	9.463	4.176
Square feet of heating surface per I.H.P. (total)	3.734	8.461
Pounds coal per I.H.P. per hour (main engines only)	1.761	2.395
Pounds coal per square foot of grate per hour	16.67	10.00
Kind and quality of coal	Pocahontas, run of the mine.	
Average pounds coal used per hour	1,800.00	1,080.00
Cooling surface, main condenser, sq. ft. per I.H.P., main engine only	1.08	2.454
Slip of propeller in per centum of its own speed	13.057	10.951

SALT-WATER EVAPORATORS.

CALCULATION OF BLOW-DOWN LOSS, HEAT BALANCE, ETC.

By WM. L. DEBAUFRE, MECHANICAL ENGINEER, ASSOCIATE.

While the blow-down loss in operating evaporators for the production of fresh water on shipboard is only a small proportion of the total heat energy imparted to the brine to produce evaporation, it is of interest to know how this loss varies with the method of operation. The effects of the rate of evaporation, of the quantity and concentration of the brine contained in the evaporator, of the method of discharging this brine, and of the degree of salinity of the feed, are all shown by the formula derived below. This formula was derived primarily to determine the relative blow-down losses of an evaporator with normal sea-water feed having a salinity of $\frac{1}{32}$ and with feed having a salinity of about one-fourth of this amount, as obtains with Severn River water used in evaporator tests at the Naval Engineering Experiment Station, Annapolis, Md.

Before proceeding further, it is necessary to settle upon a proper definition of salinity to serve as a basis for the derivation of the formula for blow-down loss. Salinity is therefore defined as the ratio of the weight of solid matter to the weight of pure water in a brine. Thus, a brine of salinity $\frac{1}{32}$ would be formed by adding 1 pound of salt to 32 pounds of pure water, making 33 pounds of brine. Salinity should not be confused with specific gravity. The specific gravity of the above brine will not be $1\frac{1}{32}$ because the volume of the 33 pounds of brine will be greater than the volume of the 32 pounds of pure water to which the 1 pound of salt is added; consequently, the density will be less than $1\frac{1}{32}$.

The following notation will be used as illustrated in Fig. 1:

Let w = weight in pounds of pure water contained in the brine in the evaporator,

k = mean salinity of brine in 32ds,

n = variation from mean value of salinity of brine in 32ds,

f = salinity of feed in 32ds, and

E = rate of evaporation of fresh water in pounds per hour.

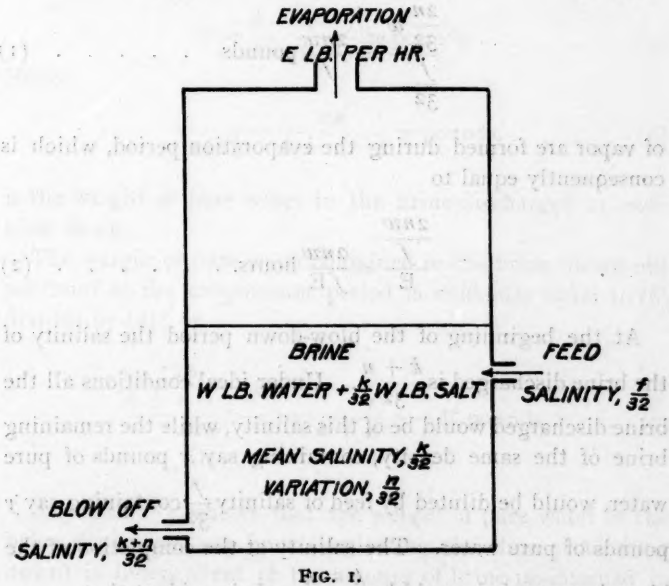


FIG. 1.

The maximum salinity of the brine in the evaporator reached just before blowing down will therefore be $\frac{k+n}{32}$. The minimum salinity just after blowing down will be $\frac{k-n}{32}$. During the period of evaporation the salinity will increase from $\frac{k-n}{32}$ to $\frac{k+n}{32}$.

The above increase in salinity of $\frac{2n}{32}$ corresponds to an increase of $\frac{2n}{32}w$ pounds of salt to the w pounds of pure water in the brine present in the evaporator, which fact becomes evident by referring to the definition of salinity. This salt comes from the evaporation of a certain number of pounds of pure water which originally entered as feed of salinity $\frac{f}{32}$. That is,

$$\frac{\frac{2n}{32}w}{\frac{f}{32}} = \frac{2nw}{f} \text{ pounds} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

of vapor are formed during the evaporation period, which is consequently equal to

$$\frac{\frac{2nw}{f}}{E} = \frac{2nw}{fE} \text{ hours.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

At the beginning of the blow-down period the salinity of the brine discharged is $\frac{k+n}{32}$. Under ideal conditions all the brine discharged would be of this salinity, while the remaining brine of the same density, containing say x pounds of pure water, would be diluted by feed of salinity $\frac{f}{32}$ containing say y pounds of pure water. The salinity at the completion of the blow-down period would be $\frac{k-n}{32}$. It is evident that,

$$x + y = w. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The amount of salt to the w pounds of pure water in the evaporator is $\frac{k-n}{32}w$ pounds at the completion of the blow down. This is made up of the salt in the brine of salinity

$\frac{k+n}{32}$ not blown out, $\frac{k+n}{32} \times x$ pounds, and of the salt in the make up feed, $\frac{f}{32} \times y$ pounds; that is,

$$\frac{k+n}{32} \times x + \frac{f}{32} y = \frac{k-n}{32} w. \quad . \quad . \quad . \quad . \quad (4)$$

Combining equations (3) and (4), we obtain for the weight of pure water in the brine not blown out,

$$x = \frac{k-n-f}{k+n-f} w \text{ pounds}. \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Hence,

$$w - x = \frac{2n}{k+n-f} w \text{ pounds}. \quad . \quad . \quad . \quad . \quad (6)$$

is the weight of pure water in the brine discharged at each blow down.

The weight of pure water contained in the brine blown out per hour of the evaporation period is evidently equal to (6) divided by (2); or,

$$\frac{\frac{2n}{k+n-f} w}{\frac{2nw}{f\,\mathrm{E}}} = \frac{f}{k+n-f} \mathrm{E} \text{ pounds}. \quad . \quad . \quad (7)$$

Expression (7) shows that the weight of pure water in the brine discharged per hour (neglecting the time of blowing down) is independent of the amount of brine maintained in the evaporator (since w cancels out of numerator and denominator), is directly proportional to the rate of evaporation E, increases with higher degrees of salinity f in the feed but at a more rapid rate, decreases with a higher mean salinity k maintained in the evaporator brine, and is less the greater the variation n permitted in the brine density. The above facts are brought out in the curves, Fig. 2 to Fig. 5.

The curves in Fig. 2 show how the weight of pure water in

the brine discharged per pound of fresh water evaporated varies with the salinity of the feed for several ranges of the salinity of the brine within the evaporator from a mean value of 2 thirty-seconds. The curves in Figs. 3, 4 and 5 show the

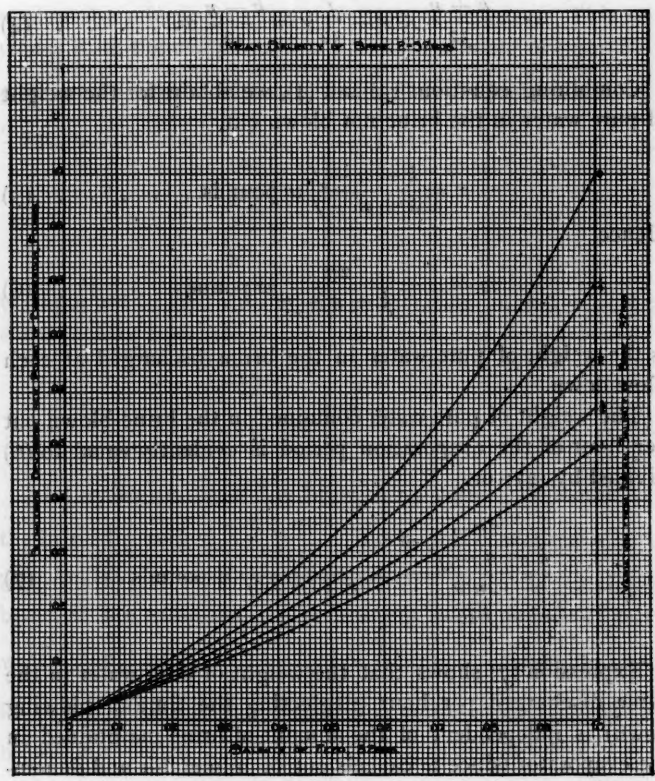


FIG. 2.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS RANGES IN SALINITY, THE MEAN SALINITY BEING 2 THIRTY-SECONDS.

corresponding variation when the mean salinities are 2.5, 3, and 4, respectively. It will be noted that the greatest discharge occurs in each case when the variation in salinity is zero, that is, for operation with a continuous blow. The

apparent conclusion ought not be at once drawn, however, that evaporators should not be operated with continuous blow.

To further investigate the question of continuous *versus* intermittent blow, the curves in Fig. 6 were plotted. These

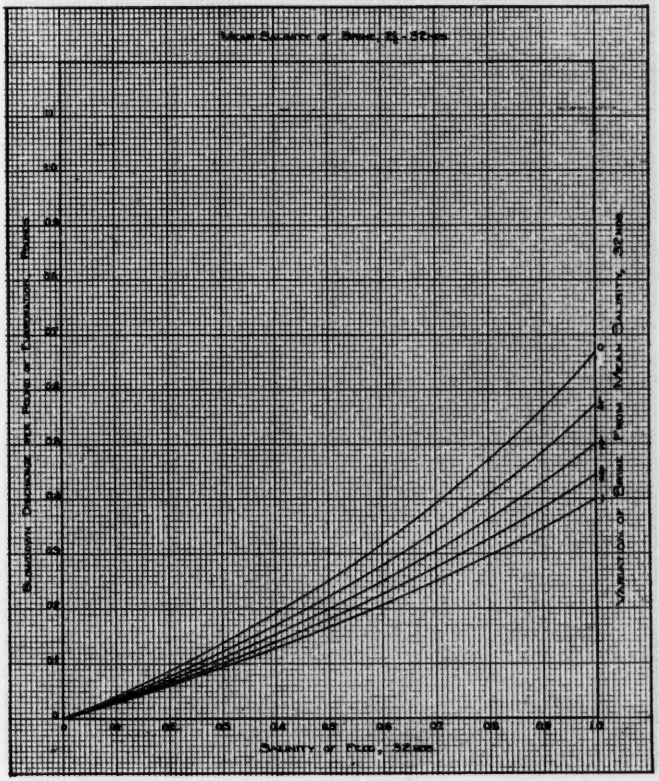


FIG. 3.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS RANGES IN SALINITY, THE MEAN SALINITY OF THE BRINE BEING 2½ THIRTY-SECONDS.

curves show the variation in the weight of pure water in the brine discharged per pound of fresh water evaporated, with the range of variation of salinity of the brine within the evaporator from the mean value and for several mean salinities,

the salinity of the feed being that of sea water, one thirty-second. The loss increases as the range of variation decreases, being greatest for no variation from the mean, corresponding to continuous blow. When no variation from the mean is

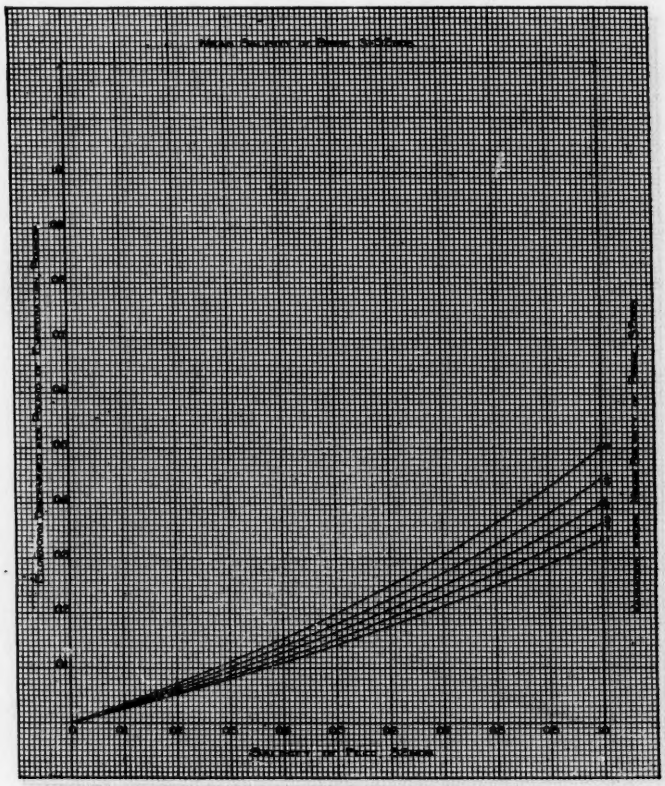


FIG. 4.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS RANGES IN SALINITY, THE MEAN SALINITY OF THE BRINE BEING 3 THIRTY-SECONDS.

permitted, the maximum salinity evidently corresponds to the mean value; when variation is permitted, the maximum salinity is higher than the mean value by the range of the variation permitted from the mean value. It is now desired to

learn how the discharge varies with the range from the mean value for any given maximum salinity.

Assume a maximum salinity of 3 thirty-seconds; from Fig. 6 the discharge with continuous blow is 0.5 pound of pure

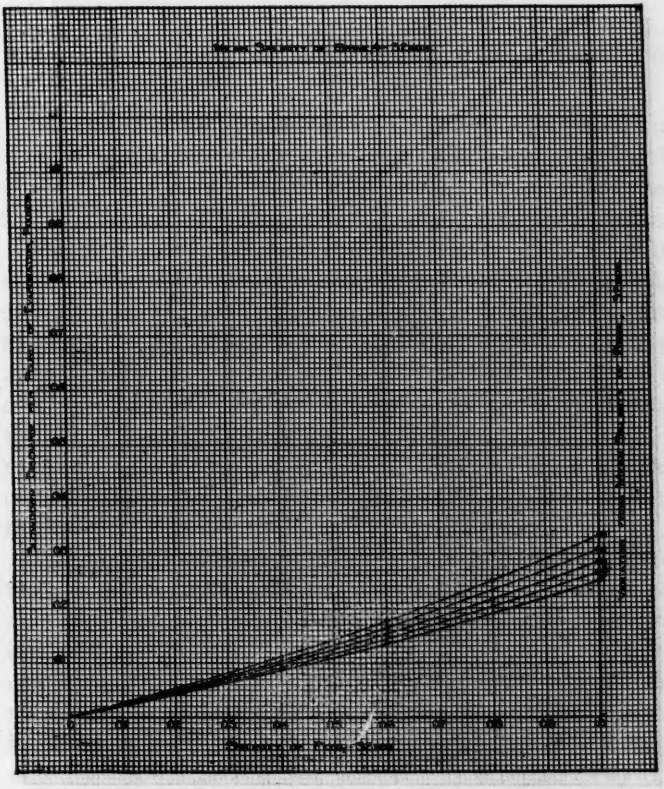


FIG. 5.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS RANGES IN SALINITY, THE MEAN SALINITY OF THE BRINE BEING 4 THIRTY-SECONDS.

water in the brine per pound of evaporation. If a range of 0.5 thirty-second from the mean is permitted, a maximum of 3 thirty-seconds corresponds to a mean salinity of 2.5 thirty-seconds; and for this case also Fig. 6 shows a discharge of

0.5 pound per pound of evaporation. The same discharge, as indicated by three curves, results from a maximum salinity of 3 thirty-seconds with any corresponding mean salinity and range. That is, the weight of pure water in the brine dis-

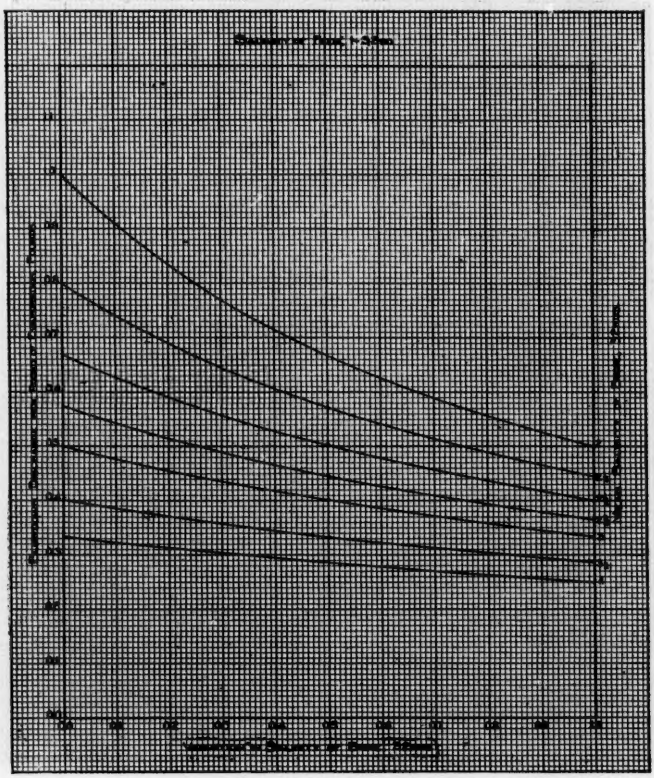


FIG. 6.—VARIATION OF BLOW-DOWN DISCHARGE WITH VARIATION OF SALINITY FROM MEAN VALUE FOR VARIOUS MEAN BRINE SALINITIES.

charge per pound of evaporation depends upon the maximum concentration attained in the evaporator and is independent of whether the blow down is continuous or intermittent.

The above fact might have been derived directly from

formula (7). Thus, instead of considering k and n as separate quantities, their sum might be replaced by m = maximum permitted salinity of brine in 32ds. Then, the weight of pure water in the brine discharged per pound of fresh water evaporated is equal to

$$M = \frac{f}{m - f} \text{ pounds.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The curves in Fig. 7 and Fig. 8 are plotted for the variation of the weight of pure water in the brine discharged per pound of evaporation of fresh water with the maximum salinity m of the brine and with the salinity f of the feed, respectively.

The former, Fig. 7, shows the advisability of working at as high a maximum brine concentration as permissible, which concentration will be limited by the greater tendency of the evaporator to prime as the concentration increases. If the impurities in any part of the fresh water produced by the evaporator are limited to a certain number of grains per gallon, then it is immaterial whether the blow be intermittent or continuous, provided the maximum concentration is not exceeded at which the impurity of the fresh water is just within the desired limit.

If, however, the average impurity only is limited, or if the fresh water produced is used for two purposes in one of which a greater impurity is permissible than in the other, then the intermittent blow results in less loss. For, if the blow were continuous, the maximum concentration of the brine could not exceed that corresponding to the average impurity; while if the blow were intermittent, the maximum concentration might be higher because the greater impurity caused by the high concentration just before the blow down would be diluted by the purer water formed at the low concentration just after the blow down. Also, with intermittent blow, the fresh water produced just after blow down could be used for one purpose where very pure water is required, while the less pure water formed later could be used for the other purpose where less purity would be permissible.

The above conclusions are based upon the relative blow-down losses under the several conditions; but the total heat loss, as will be shown later, is a comparatively small proportion of the heat interchange involved. Consequently, these

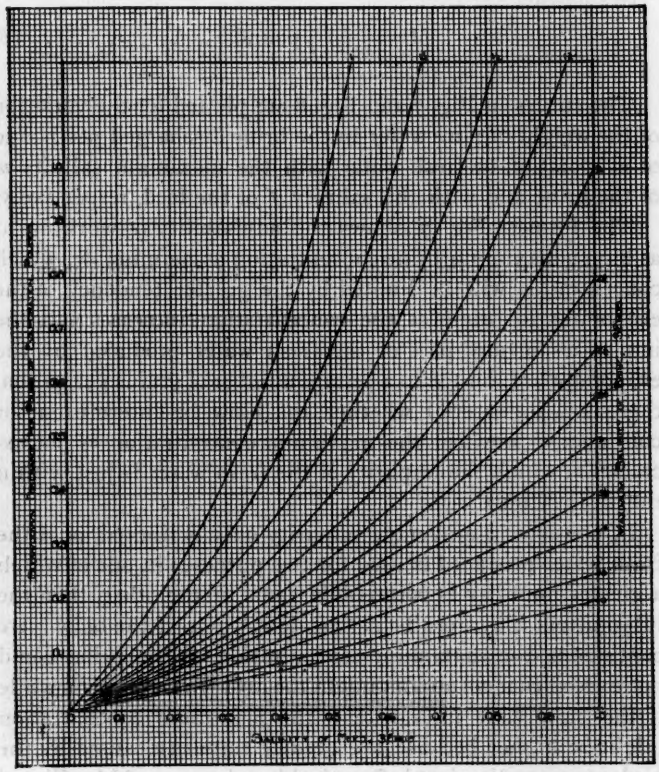


FIG. 7.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS BRINE CONCENTRATIONS.

conclusions may be modified by practical operating conditions. The main point is that the least loss occurs when the greatest maximum concentration of the brine considered allowable under the operating conditions is reached, and the loss is to

that extent independent of whether the blow be continuous or intermittent.

The curves in Fig. 8 are of greatest use when it is desired to calculate the blow-down discharge for a feed salinity different

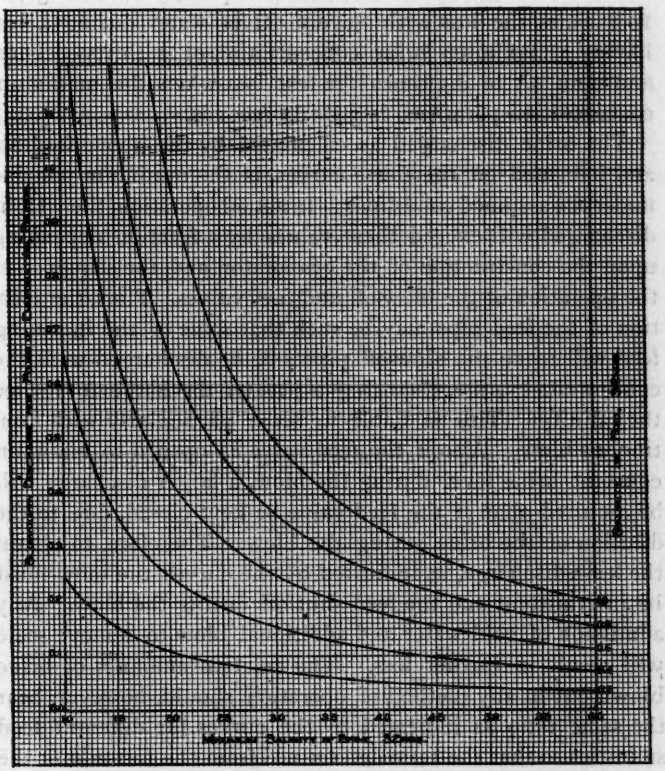


FIG. 8.—VARIATION OF BLOW-DOWN DISCHARGE WITH BRINE CONCENTRATION FOR VARIOUS SALINITIES OF FEED.

from that at which a test was made of an evaporator. It is evident that the feed salinity and the maximum brine concentration should be measured during the test. In the Severn River water available at the Naval Enginering Experiment

Station for testing evaporators there are from 200 to 350 grains of chlorine per gallon. At sea water of salinity 1 thirty-second contains about 1,200 grains per gallon, the salinity of Severn River water varies from 0.17 to 0.29 thirty-seconds, and the loss with sea water will consequently be from 5.9 to 10.8 times the loss with Severn River water during a test having a maximum brine concentration of 2 thirty-seconds, and from 4.7 to 8.3 times the test loss when the maximum concentration permitted is 3 thirty-seconds.

The brine discharged as calculated by the above formula and indicated on the above curves, is the minimum possible under the assumed conditions. In actual operation, the weight discharged will exceed these values by a fraction depending upon the construction of the evaporator and the sequence of the operation of the blow-off and feed valves. Thus, the location of the feed pipe may be such that some of the incoming feed will reach the blow-off pipe and be discharged with the concentrated brine, requiring a greater discharge to reduce the salinity. This is liable to occur particularly with continuous blow. Also if the feed valve is opened to maintain constant brine level during the blow, a larger discharge will be required than if the feed valve be kept closed, due to the dilution of the brine during the blow. Therefore, to obtain the least discharge, as indicated by the curves, the blow must be intermittent and the feed shut off during the blow. Any admission of cold water and additional blow to remove scale will result in further loss. The relative amounts of brine blown out in any method of operation should, however, bear the same ratio as indicated by the curves for different feed salinities so that the ratio determined from the formula or the curves can be applied to reduce the conditions with a measured feed salinity to those corresponding to a standard value.

The heat loss will equal the product of the weight of brine discharged, its specific heat, and some temperature difference, the upper limit of which will be the temperature of the brine discharged. This should nearly correspond to the vapor pressure maintained in the evaporator, a much less temperature

indicating dilution by cool feed. The fact that when the feed short circuits to the blow-off pipe, it is not heated to the concentrated brine temperature, shows that the blow-down loss from faulty construction is not as much as might be expected. Consequently, it is considered sufficiently accurate to calculate the weight by the formula proposed and assume the discharge temperature to be that corresponding to the vapor pressure as taken from steam tables, the temperature of boiling fresh water being slightly less than boiling brine for the same vapor pressure.

The correct lower temperature to assume will depend upon whether the efficiency of the evaporator alone or of the whole evaporating plant is under consideration. In the former case, the correct temperature will be that of the feed to the evaporator. In the latter case, it will be the temperature of the sea water available.

The values in the first two columns of Table I below are taken from "Marine Boiler Management and Construction" by C. E. Stromeyer.

TABLE I.—SPECIFIC HEAT OF SALT WATER.

Weight of salt added to 1 pound water, pound.	Specific heat of salt water, B.t.u. per pound brine.	Weight of brine containing 1 lb. pure water, lbs.	Specific heat of salt water, B.t.u. per lb. pure water.
1	2	3	4
.0	1.000	1.000	1.000
0.016	0.978	1.016	0.9936
0.049	0.945	1.049	0.9875
0.115	0.877	1.115	0.9779
0.123	0.871	1.123	0.9781
0.243	0.791	1.243	0.9832

The fourth column is the product of the second and third columns. The slight variation of the specific heat of salt water based upon one pound of pure fresh water contained therein indicates that no modification need be made of the foregoing conclusions based on the weight of pure water in the brine

discharged. For most calculations the value of this "specific heat" may be taken as unity.

In the heat balance of an evaporator considered apart from the complete water-distilling system, the only losses are the heat in the blow-down discharge and the heat radiated. The latter quantity is negligible with a well-lagged evaporator. The heat in the condensate from the steam supplied to evaporate the brine is not considered a loss when dealing with the evaporator alone, for this condensate is available as feed water to a boiler which then has to supply the latent heat only. All this latent heat is imparted to the brine to evaporate it. Consequently the above formula and curves will serve to calculate the efficiency in any given case. For example: Required the efficiency of an evaporator in which the shell pressure is maintained at 30 pounds per square inch gage, the salinity of the feed being 1 thirty-second and the maximum brine concentration 2.5 thirty-seconds. Assume a feed temperature of 70 degrees F. Now, corresponding to 30 pounds gage, the absolute pressure is 44.7 pounds per square inch, the corresponding steam temperature 274.0 degrees F., at which the latent heat is 928.6 B.t.u. per pound and the total heat is 1,171.5 B.t.u. per pound. The sensible heat at 70 degrees F. is 38.1 B.t.u. per pound. The formula or curves show that two-thirds pound of pure water will be contained in the brine blown out per pound of fresh water evaporated. There will thus be $\frac{2}{3} \times 1 \times (274 - 70) = 136.0$ B.t.u. in the brine discharged, and $1{,}171.5 - 38.1 = 1{,}133.4$ B.t.u. will be required to evaporate one pound of fresh water, necessitating a supply of $1{,}133.4 + 136 = 1{,}269.4$ B.t.u. Consequently, the efficiency of the evaporator will be $\frac{1{,}133.4}{1{,}269.4} = 0.8929$ or 89.29 per cent.

If the steam in the coils was maintained at 50 pounds gage during the evaporation, the ratio of water evaporated to steam condensed would be determined as follows: The latent heat of steam at 50 pounds gage, or 64.7 pounds per square inch absolute, is 911.2 B.t.u. per pound. Therefore, $\frac{911.2}{1269.4} = 0.7178$

pound of vapor is produced per pound of steam. It must be noted that the above calculations will be modified by moisture or superheat in the steam supplied and by superheat in the vapor produced. There should be no appreciable moisture in the vapor, for such would mean impure water.

Since the efficiency of an evaporator is so easily calculated under any assumed conditions, a test of a given evaporator is not conducted primarily to determine its efficiency but to determine its capacity and whether it will produce water of sufficient purity. Evaporator tests may also be conducted to determine the effects upon capacity and purity of various modifications in construction or operation. The efficiency might be affected by short circuiting some of the feed to the blow-off connection, but this effect should be slight because this short-circuited feed will not have been warmed to the evaporator temperature. In the above discussion, m could have been called the salinity of the discharge rather than the maximum salinity of the brine in the evaporator. This was avoided, however, to insure a definite understanding of the upper temperature to be used in calculating the heat lost in the discharge.

The upper temperature corresponds to the vapor pressure in the evaporator shell. Any feed short circuiting to the blow-off connection to increase the discharge does not increase the heat loss, assuming it to receive no heat while passing through the evaporator. The resulting calculations based upon this assumption are simpler than considering m to be the salinity of the discharge and then calculating the temperature resulting from the mixture of cool feed with the warm brine.

The formula and calculations have also been maintained simpler by dealing with the weights of pure water in the various brines rather than would have resulted by adding the weights of the saline matter to obtain the total weight of brine. If the total weight is desired, however, it can be easily obtained; thus, the total weight of brine discharged per pound of fresh water produced is $\frac{f}{m-f}\left(1+\frac{m}{32}\right)$.

SUMMARY.

The blow-down discharge of an evaporator may be calculated by the formula,

$$M = \frac{f}{m - f},$$

in which M = weight of pure water in the brine discharged per pound of fresh water evaporated, in pounds,
f = salinity of the feed, in 32ds, and
m = maximum salinity attained in the evaporator, in 32ds.

The salinity of the discharge will be m unless some of the feed short circuits to the blow-off connection and increases the amount of the discharge.

As the short-circuited feed will not with proper construction have sufficient time to absorb much heat, the heat lost in the discharge will equal,

$$H_m = M \times (t_v - t_f),$$

in which H_m = heat loss per pound of fresh water evaporated, in B.t.u.,
M = weight of pure water in the brine discharged per pound of fresh water evaporated, in pounds,
t_v = temperature corresponding to the vapor pressure in the shell, in degrees F., and
t_f = temperature of the feed, in degrees F.

The "specific heat" to raise the temperature of brine containing one pound of pure water one degree F., has been taken as unity.

The efficiency of the evaporator, neglecting the small loss by radiation, is given by,

$$e = \frac{H_v}{H_v + H_m},$$

in which H_m = value as calculated above, and

H_v = heat required to raise one pound of fresh water from the feed temperature to the temperature corresponding to the vapor pressure and evaporate it at that temperature, in B.t.u.

The above formulae indicate that the efficiency of an evaporator, considered apart from the whole distilling system, depends upon the salinity of the feed and the concentration attained in the evaporator. It is independent of whether the blow be continuous or intermittent, except in so far as this affects the short circuiting to the blow-off connection of feed which has been somewhat heated in passing through the evaporator.

Since the purity of the fresh water produced is dependent upon the average salinity of the brine in the evaporator while the loss in the blow down is dependent upon the maximum salinity, an intermittent blow down is a more economical method of operation than continuous blow. This is evident when we consider that the maximum salinity is equal to the mean salinity with continuous blow but is greater with intermittent blow.

The pounds of fresh water produced per pound of steam condensed is equal to

$$\frac{H}{H_v + H_m}$$

where H = heat given up by the steam in condensing, measured in B.t.u. per pound, and

H_v and H_m have the same significance as above.

DESCRIPTION AND TRIALS OF U. S. TORPEDO-BOAT DESTROYER *WINSLOW*.

By Lieutenant W. F. Cochrane, U. S. N., Member.

The *Winslow* is one of six destroyers authorized by an Act of Congress approved August 22, 1912, these vessels being designated as Torpedo-Boat Destroyers Nos. 51 to 56, inclusive.

The contract for building three of these boats, Nos. 51, 52 and 53, named *O'Brien*, *Nicholson* and *Winslow*, was awarded to the Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Pa., and was signed December 7, 1912; the time allowed for completion being 23 months for the first boat, 23½ months for the second, and 24 months for the third.

The designed speed was 29 knots, at about 1,050 tons displacement.

The contract price for each vessel was $842,000.00, of which $502,000.00 was allotted for machinery.

The dimensions of vessel, machinery details, etc., are the same as those given in Journal of the American Society of Naval Engineers, Volume XXVII, No. 2, of May, 1915.

TRIALS.

The contract required:

(*a*) A progressive trial over the measured-mile course at Lewes, Del., for standardizing the screws, extending from maximum speed (at least 29 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 29 knots. The speed to be determined by the average revolutions of the main shafts, according to the official standardization curve.

(*c*) A fuel-oil and water-consumption trial of four hours'

duration in open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service cruising conditions.

(*d*) A fuel-oil and water-consumption trial of four hours' duration at 15½ knots, under conditions similar to the preceding trial, but with the cruising engines connected and in use.

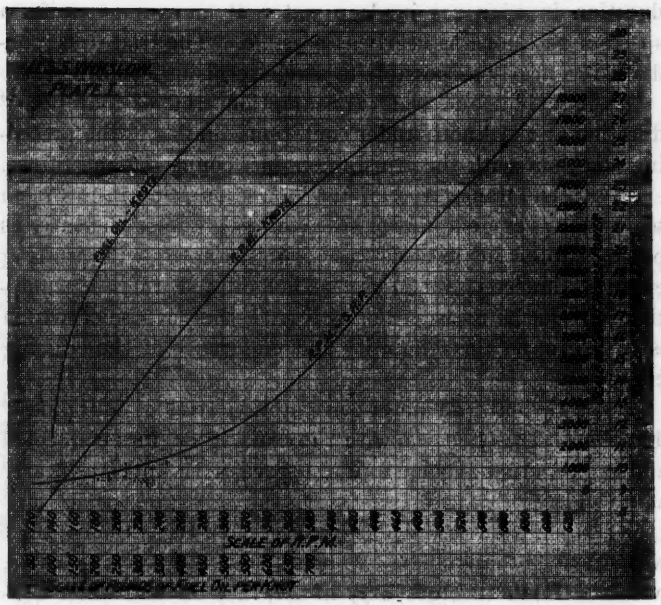
(*e*) An endurance trial of ten hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engines connected and in use. Fuel oil and water consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engines.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea, with the cruising engines connected and in use, at an average uniform speed of 12 knots, as nearly as possible.

(*g*) In addition to the above-enumerated trials the contract was amended to include a two-hours' trial at about 15½ knots, with the main turbines only in use, fuel oil and water consumption to be carefully measured on this trial.

Fuel-Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 692.25 pounds at guaranteed maximum speed, 444.6 pounds at 24 knots, 213 pounds at 15½ knots, and 170 pounds at 12 knots, the consumption of the fuel oil at these speeds to be determined by the Trial Board from a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*), and corrected to a standard of 19,500 B.t.u. per pound of fuel oil.

Standardization Trial (*a*).—This trial was conducted under favorable weather conditions on the measured mile at Lewes, Delaware, on June 29, 1915. Twenty-eight runs were made over the course at various speeds, and from the data obtained the revolution, speed and power curves in Plate I were plotted. Table I gives the standardization data.



From the curve in Plate I the following r.p.m. of the propellers were found to be necessary for the various speeds:

12 knots, ...	192.70
15½ knots, ...	249.20
24 knots, ...	413.40
29 knots, ...	571.20

Four-Hour 12-Knot Fuel-Oil and Water-Consumption Trial (*f*).—This trial commenced at 1:32 P. M., June 29, 1915, and was completed at 5:32 P. M., the same day. The weather was fair and the trial very successful. For data see Table II.

Four-Hour Full-Speed Trial (*b*).—This trial began at 7:55 A. M., June 30, 1915, off Five-Fathom Bank Lightship, and was completed at 11:55 the same day. The weather was fair. The data obtained on this run is given in Table II. The machinery operated excellently. All guarantees were easily attained.

TABLE I.—STANDARDIZATION U. S. S. "WINSLOW," LEWES, DEL., JUNE 29, 1915.

No. of run.	Speed, knots.	R.P.M.			S.H.P. Mean.	Speed Ave. Group.	S.H.P. Ave. Group.	R.P.M. Ave. Group.
		Stbd.	Port.	Ave.				
1	8.46	131.49	130.29	130.89	234.00			
2	7.32	130.82	130.69	130.76	273.00	8.11	259	131.26
3	9.32	132.92	132.35	132.64	257.00			
4	11.39	200.77	199.99	200.38	797.00			
5	13.45	198.03	198.90	198.47	770.00	12.36	782	198.44
6	11.16	195.68	197.20	196.44	792.00			
7	17.41	253.49	254.07	253.78	1,540.00			
8	13.91	253.25	253.06	253.16	1,523.00	15.75	1,537	253.29
9	17.78	253.22	252.88	253.05	1,560.00			
10	18.33	330.81	330.58	330.70	3,453.00			
11	22.17	331.04	329.78	330.41	3,450.00	20.26	3,455	330.15
12	18.35	329.35	328.80	329.08	3,468.00			
13	25.82	419.74	416.54	418.14	7,566.00			
14	22.54	416.73	414.46	415.60	7,459.00	24.10	7,503	416.04
15	25.51	416.81	412.81	414.81	7,527.00			
16	24.67	470.07	467.19	468.63	10,763.00			
17	27.21	467.23	466.12	466.68	10,580.00	25.94	10,598	466.72
18	24.66	464.79	464.94	464.87	10,470.00			
19	29.73	565.83	562.61	564.22	15,904.00			
20	27.89	561.57	561.94	561.76	15,642.00	28.79	15,782	563.79
21	29.63	568.83	565.97	567.40	15,938.00			
22	29.01	594.53	594.48	594.51	17,558.00			
23	30.18	597.67	594.85	596.26	17,490.00			
24	29.53	598.85	604.14	601.50	17,915.00	29.85	17,828	600.86
25	30.23	607.94	603.65	605.80	18,221.00			
26	29.90	602.87	607.60	605.24	17,813.00			

Four-Hour 24-Knot Fuel-Oil and Water-Consumption Trial (*c*).—Following the full-speed trial the 24-knot trial began at 1:45 P. M., and was completed at 5:45 P. M. The weather was fair. The trial was very successful. For data see Table II.

Two-Hour 15½-Knot Fuel-Oil and Water-Consumption Trial (*g*).—The trial began at 6:30 P. M., following the 24-knot run, and was completed at 8:30 P. M. The trial was very successful. Data is given in Table II.

Four-Hour 15½-Knot Fuel-Oil and Water-Consumption Trial (*d*).—The trial began at 6:30 A. M., July 1, 1915, and was completed at 10:30 A. M., the same day. The trial was very successful. The weather was fair. For data see Table II.

Ten-Hour 15½-Knot Endurance Trial (e).—This trial began at 11:00 A. M., July 1, 1915, and was completed at 9.00 P. M., July 1, 1915. The weather was fair. The trial was very successful, and the reciprocating engines and clutches operated excellently.

TABLE II.—U. S. S. "WINSLOW."

.....	4-hour full power.	4-hour 24-knot.	4-hour 15½-knot.	4-hour 12-knot.	2-hour 15½-knot.
Date of trial	6-30-15	6-30-15	7-1-15	6-29-15	6-30-15
Displacement	1,041	1,042	1,046	1,047	1,057.5
Boilers in use	4	4	2	2	2
Heating surface	21,600	21,600	10,800	10,800	10,800
Speed in knots	29.054	24.036	15.560	12.08	15.567
R.p.m., starboard	573.21	414.30	250.35	193.69	250.55
port	573.09	414.30	250.19	194.34	250.18
mean	573.15	414.00	250.27	194.02	250.37
S.H.P., mean	15,984	7,384.00	1,593.0	672	1,524
I.H.P., cruising engines	.....	.....	.....	.....	.....
Pressures:					
Main steam (G)	243.69	234.69	246.0	225	238
Full speed (abs.) S.	211.44	75.00	14.8	9.69	20
Full speed (abs.) P.	215.00	78.88	14.8	9.56	25
Cruising speed (abs.) S.	157.94	207.25	40.63	23.13	65
Cruising speed (abs.) P.	157.18	201.13	34.63	20.00	65
10th stage (abs.) S.	26.84	12.63	3.56	2.0	6.5
10th stage (abs.) P.	26.84	13.31	1.06	1.5	4.5
14th stage (abs.) S.	14.0	6.8	1.93	1.31	2.5
14th stage (abs.) P.	13.25	6.0	1.50	1.0	2.0
Gland steam (G)	6.25	5.19	4.13	2.57	4.94
Cruising engines:					
H.P. valve chest	.....	.....	238.32	144	.....
L.P. receiver	.....	.....	42.57	21.8	.....
Vacuum, starboard	27.2	27.95	28.05	28.5	28.3
port	27.4	28.10	27.99	28.1	28.5
Auxiliary exhaust	8.87	8.38	7.5	3.6	6.25
Lub. oil to main engines	14.13	12.18	11.19	11.5	10.13
Oil to clutches	.....	.....	76.25	75.5	.....
Temperatures:					
Air	74	73	80	74	70
Engine room	79	78	87	82	79
Auxiliary room	84	83	82	81	82
Fireroom	96	95	104	101	103
Oil to cooler	112	103	100	94	94
from cooler	90	58	90	86	86
Main injection	66	66	69	66	66
discharge	92	83	78	71	76
Fuel oil to heaters	73	74	88	.....	78
from heaters	76	79	88	.....	88
Smoke pipes	550	390	280	245	415
R.P.M. blowers	1,510	1,350 (2)	746 (1)	621 (1)	1,170 (1)
Air pressure	6.75	3.70	1.86	1.46	3.08
Water per S.H.P., M.E.	16.158	18.244	22.435	25.623	31.33
Water evap. per lb. oil	13.614	14.66	15.38	14.775	7.463
Fuel oil per knot run	661.50	388.0	138.0	105.0	199.0

OIL BURNING.

By A. M. R. Allen, Lieut., J. G., U. S. N., Member.

REFERENCES:

Journal American Society Naval Engineers,
Vol. XXIII, No. 2, "Oil Fuel," by Peabody.
Vol. XXIV, No. 4, "Tests of Blower Fans."
Vol. XXIII, No. 1, "Notes on the Burning of Liquid Fuel," by Lovekin.
Viscosity Curves, from Bureau.
Carborundum Refactories, A pamphlet from the Carborundum Company, Niagara Falls, N. Y.

In taking up a new installation, the first thing to determine is the effectiveness of the pumps and heaters. Oil lines are tested to 600 pounds' pressure, and, in starting up, it is best to test everything, preferably with water, to see that all joints on lines and heaters, and all cocks and valves are tight. Try the line out in sections, taking care not to exceed this high pressure on any one section. Make notes of all leaks. Take particular care and see that the oil heater is tight. All modern heaters have outside packed joints, so that oil cannot get into the steam drainage. The Schutte-Koerting heaters have plugs so arranged that you can easily tell if there is any leakage from the steam to the oil side. Each installation has its necessary gages and test cocks arranged for just such testing, and, if not, it will be well to arrange for the installation of any test instruments and openings necessary in the very beginning, while still in port. The air flasks on the line, and all joints and valves connected with them, must be given particular attention.

After you have thoroughly tested and know that the oil line and valves are tight up to the burners, the next point to study is the burner spray. By testing with water and trying dif-

ferent pressures, one, or a certain range, will be found which will give the clearest spray. In doing this great care must be taken to see that the test burner, particularly the tip, is perfectly clean, or otherwise results will be very unreliable. The writer found that about 200 pounds pressure while steaming, and 150 pounds for port use gave very satisfactory results with the Schutte-Koerting system; while with the Ingram burner, using steam, about 40 pounds of oil pressure would give excellent results, as long as the burner was clean. As the burners get dirty, which shows by the increased pressure in the Ingram, and by streaks in the oil cone in the Schutte-Koerting and Bureau types, they must be removed and cleaned.

After a satisfactory spray pressure has been found, the next step is to so adjust the burner, if adjustment is possible, that the cone of oil just clears the brick rings in the furnace. These points are sometimes cared for in design, but in many cases, where Bureau burners have been substituted, the question of adjustment is entirely up to the engineer officer.

After the burners are right the next question is lighting off. Try out the hand pump and see what is the maximum pressure you can make with it. Open the line to the burners and stand by. In the meantime, see that the smoke-pipe cover is off, ventilator rigged, water at steaming level, auxiliary feed pump ready for use, and fuel-oil pump ready; air cock open, all gage glasses working, feed stops and checks working, bottom blows tightly closed, main and auxiliary stops closed, and safety valves and easing gear in working order. The last thing, look into the fire box through the registers and make sure no oil has collected in the furnace and that none of the burners are dripping. Prepare a long ⅜-inch iron rod, with a hook in one end to hold some waste for a torch, and put a little oil on it. Seeing the pressure on the line, light the torch and stick it in through the register, shutting the latter on it. Crack the burner valve, shutting off the valve if the oil does not ignite at once. This is the time for greatest care, for if an inexperienced man allows the furnace to fill with heavy

oil gas, which will not rise owing to the lack of draft, a heavy explosion will take place, injuring both the man and the boiler. It may be necessary, if in lighting off the first time the torch burns out before the burner lights properly, to get an air or steam hose from the deck to induce a good draft before again putting a lighted torch through the register. One point in particular is to be emphasized, and that is, that the man handling the torch must keep his face away from the register. The natural tendency is to get down close and see what sort of a stream the burner throws, but this must not be done. The only oil accident during the time I was engineer officer was due to a man doing this, and, fortunately for him, it only resulted in severe burns.

After the burner lights off properly the hand pump must be kept going until there is steam enough to run the pumps; then they must be started and the burners cut in as fast as possible, steam being also put into the heaters, if necessary. Oil, of course, becomes less viscous under heat, and flows more easily, and the point is to give just sufficient heat to make the oil have a viscosity of about 8 Engler and no more, for more simply wastes heat units.

Most destroyer installations will work well without a blower for port use, unless the evaporators are in use or an excessive amount of electricity is being used. This is best judged by the fireman from the appearance of his fire, and the engineer by the look of his stack. A light gray haze is the ideal condition, for the reason that it indicates complete combustion, giving a stack temperature of about 500 degrees on the pyrometer. Too little air causes excessive black smoke and panting, and too much air causes white smoke, which means that the excessive amount of air is combining with the small amount of oil and forming steam which is condensed when it reaches the air.

Panting, that condition of the boilers due to excessive amount of oil—more than the air supplied, can burn at the velocity at which it is being fed to the boiler, is the most in-

jurious factor in oil burning. It is a condition that should never be allowed to continue, for it shakes down the brickwork, loosens up the joints in the casing, and racks the boilers, connections, and ship itself in a highly injurious manner. It can be overcome by increasing the velocity of air, by slightly closing the registers, or by cutting out burners, or by increasing the air pressure by speeding up the blowers. Within limits the latter is the quickest and most effective measure, and the one chiefly employed in maneuvering. In this connection, it may be said that putting both blowers on a common throttle has tended greatly to increase the efficiency of a fireroom with two blowers, as it insures their being speeded up together, and in so doing prevents the increased pressure escaping through the slow burner, as was the condition before this was done. Panting is also frequently caused by insufficient air in the banks, which should always be charged as soon as there is steam on the ship. If air from the torpedo charger is used through a reducer, they can be charged at full working pressure, but if the Westinghouse compressor is used the oil pressure may have to be slightly lowered until through charging. The air banks, or "pigs" as they are called, should show a half a glass of air at working pressure, and must be pumped up frequently, as the oil absorbs the air. The pressure gage on the oil line should show a steady pressure regardless of the pump strokes, as no installation using mechanical atomization can be successfully operated unless this condition obtains.

The next test to be carried out is to test all casings, throat sheets and burner openings for tightness; also all openings into the firerooms themselves. This, of course, can best be done in a dead fireroom, and a good approximation for proper tightness is to close all openings, speed up the blowers to a fixed number of revolutions, and see what pressure can be maintained. All boilers must have some one inside to see where the air leaks into them, and all seams, stuffing boxes, doors, deadlights, etc., around the firerooms must be carefully gone over. If there appear to be any leaks, soap and water

will sometimes aid in detecting them. On the *Jarvis* the vent from C-2, the reserve-feed tank in the engine room, was open to the fireroom, and this caused an air loss and made false readings in the water tank. The blower fans should run with as little clearance as possible, otherwise there will be much leakage around them. In carrying out this test care should be taken to see that the air gage is accurately filled, and as leaks are stopped the increase of pressure noted. The point of all this is to make sure that all the air pressed into the firerooms passes out through the burner registers, and the economy lies in the fact that, for every leak stopped, the velocity necessary to keep a given number of burners going can be obtained for less revolutions of the blowers, which should be run with as little steam as possible. The fact must be borne in mind, that in free route, when sudden increase of power is not likely to be demanded, one blower run at full speed is more economical than two run slowly. To do this, however, the blower shut down must have an absolutely tight hood.

Another point to remember in blower operation is that, when using the common throttle care must be taken to adjust the independent throttle so that there will be an equal increase in speed in both blowers for a given opening of the common throttle. This is greatly facilitated by tail rods on the blowers, and deck throttles, so that a man on deck can adjust this. Once adjusted it should be frequently checked.

Testing of feed pumps, exhaust lines, feed heaters, etc., must be carried out, but these tests are not within the province of this article to consider.

The next test is a heat test, and the principle to be borne in mind is the fact that the least possible amount of steam that can be used to give the oil sufficient fluidity to burn properly is all that is required. This must be studied, as it varies with the quality of oil and the type of burner. The base of all experiments can be taken from the "Viscosity Table," gotten up at the oil-testing plant in Philadelphia, and only sufficient temperature used to give a viscosity of 8. The dis-

tance between the thermometer and the burners must also be considered, and it will take some little time to establish a set of standard conditions for any installation.

The three important tests having been completed, viz: pressure and temperature of oil, and air pressure, the next step is the establishment of standard working conditions for different speeds, and different boiler and fireroom combinations. The basis for this work should be, as far as possible, the work of the next previous destroyer from the same shipbuilding company, and the first efforts should be directed to proving out the previous results for your own installation. Unfortunately, up to the present time there has not been sufficient effort to put these results in such shape that new boats could take full advantage of the results obtained by others; and one trouble has been due to the lack of appreciation by younger engineers that only the most accurate data is of value. Too much care cannot be taken of air gages, pressure gages, thermometers and tachometers, and it must be constantly borne in mind that these instruments are the helpful sign boards that tell the engineer exactly what his installation is doing. The personnel must have this impressed upon them at all times, and must be led to realize that the care and watchfulness of the instruments on the oil system are just as important as the care of boiler gages and water glasses. They should be encouraged from the beginning to take an interest in keeping careful records of readings taken while on watch, and every effort made to show them the value of such readings. They will take interest themselves, as a rule, if you can show them clearly that the man who, while on watch, keeps everything carefully adjusted to standard conditions is the man who burns the least oil and who should get a reward for this careful work. Brains, not brawn, are required in an oil-burning fireroom.

Having established a system which will be productive of accurate data, it is the engineer's duty to so interpret this in-

formation that he can establish the *best possible standards* for all working conditions.

Underway the following method of burner adjustment must be used. The air pressure should be just sufficient to keep all burners required for a given speed going smoothly, without panting, and with a light gray haze showing from the stacks of all boilers in use. The center burners must be the first ones added, and a drop in steam pressure must be covered by immediately speeding up the blowers before cutting in more burners. Blowers must be immediately slowed when the demand for steam ceases, which is shown by the rapid rise of the gage pressure, but burners must be cut out first in this case. On several boats the engine-room annunciators have been connected to show the changes of speed in the firerooms, so that the W.T. can be prepared and handle his boiler quicker. Under ordinary working conditions the gage is the best indicator, and a properly-trained fireroom crew will hold the pressure without smoke and without popping off, except under the most unusual conditions. The W.T. has many things to watch, and if he is trained to watch his main steam gages, they are his greatest help.

One other element in training fireroom crews as separate watches and keeping them together like a gun's crew is that, for best economy, each boiler should have the same number of burners; and, when running two firerooms, this requires a good voice tube or some method of signalling between firerooms, otherwise one or the other will carry all the load instead of equally dividing it, as should be done under all conditions.

Governors on the fuel-oil pump have given general satisfaction, and they keep the oil pressure constant regardless of the number of burners in use. In maneuvering, the temperature will vary slightly due to the variation in the amount of oil passing through the heater, while the steam pressure remains constant, but this cannot be overcome in any of the installations so far installed.

Having once steadied on a given speed the standard for this speed is at once established as follows: The steam pressure must, of course, be kept constant at the highest pressure that can be carried without lifting the safety valves, and the usual method is to adjust the boilers so that each one is using the same number of burners. Then the air pressure is gradually cut down until a slight haze can be seen in the furnace through the peep, looking close under the tubes. The slightest inclination to pant is controlled by gradually closing the registers until the maximum opening for the conditions is found. The blowers are then kept steady at this speed. One boiler is assigned to carry the slight variations in steam pressure, while the other is kept constant except when necessary to clean a burner. This is shown by black streaks in the oil cone from the burner and by black smoke in the furnace. Well trained firemen will detect this at once, and, if there is a doubt as to which burner it is, will shift from one to the other until they find that one, which, when cut out, makes the furnace show clear again.

The reason that it is not possible to establish absolutely accurate standards for all fixed speeds is because of the variation in the power required to produce a fixed number of revolutions, due to change in conditions of wind, current, displacement, trim and the condition of the ship's bottom; but every effort should be made to find them for average conditions, in order to tell whether or not the speed is being made most economically.

As the revolutions are fixed, the number of burners in use is the best indication of economical working conditions; and if the number in use is excessive, investigation should at once be made to see that the vacuum, feed water, combination in use, etc., are such as to give best results. Usually, low temperature of feed water is the source of trouble. For instance, the 20-knot average working standards for the *Jarvis* are as follows: Main steam boilers, 260 pounds; engine room, 250 pounds; I.P.C., 155 pounds; M.H., 75 pounds; L.P.'s., 0;

Vacuum, 28; R.P.M., 465; back pressure, 4.5 inches; oil consumption, about 2,800 gallons a watch; and any drop in the feed temperature showed itself immediately in the number of burners required.

On ordinary runs the amount of make-up feed and oil used could be actually measured by soundings, and it was routine on all but the roughest trips to take them at the end of each watch.

The above notes will assist in getting a working basis for any installation, and the records kept should show the following:

Engine room.	P. Vac.
R.P.M. No. 1.	Back pressure.
R.P.M. No. 2.	Temperature highest bearing.
R.P.M. No. 3.	Stack temperatures.
Average R.P.M.	Temperature hotwell.
Speed.	Temperature air pumps { S
Main steam (Boiler, E. R).	discharges. { P
H.P.C.	
I.P.C.	*Firerooms.*
M.H.	Air pressure.
S.L.P.	Oil pressure.
P.L.P.	Oil temperature.
S. Vac.	Feed temperature.
(Better the temperatures of the exhausts if thermometers could be installed).	Number burners.
	R.P.M. blowers.
	Oil per watch gallons.
	Make-up feed per watch.

Some of this data can be taken from the log, and some from a sheet or book kept by the W.T. on watch, and the whole compiled in card form for ready reference. This card record should be summarized at each visit to a navy yard, from which corrections should be made in the working conditions until the ship is down to most economical standards possible to at-

tain. The number of double strokes necessary for each pump, when working under best conditions, should also be known, as this will give a check on that auxiliary, in case a loss is noted which is hard to locate.

In shutting down, the main point to remember is to keep the blowers on until the demand for steam has been reduced below the point where a single burner can take care of that required; and, if this cannot be done, keep one blower running.

The care of the installation, as regards preservation, is well covered by the regulations. Fuel oil can be used to good advantage on the steel piping to keep it from rusting. The joints in the oil line are scraped flanges made up with shellac or Eretite or other joint preparation to stand 600 pounds' test pressure, and should be most carefully cared for. Fuel-oil pumps should be packed with leather rings and kept in the best adjustment at all times. Strainers must be cleaned frequently, and the slightest indication of loss of suction or decrease in pressure due to clogged strainer must be overcome at once by shifting strainers. Every oil-burning fireroom must have a vice for holding burners while cleaning them, and a proper equipment of wrenches and reamers. Great care must be taken of burner tips, as the slightest corrosion or carbon in the tips will give trouble. A steam connection to a burner holder for blowing out burners should be furnished near the vice, with a bucket held in a bracket underneath, so that it can be taken on deck and emptied, or, better, an arrangement made for blowing it through the side of the ship. Burner cocks or valves should always be kept tight, and the ship supplied with spare discs, or, if cocks are used, with the necessary reamers and blanks for renewing the cocks if necessary.

The registers and their operating gear should be kept thoroughly clean and in such condition that, when not in use, the shutters close the openings completely and tightly. The ship should have a large piece of canvas, which can be spread com-

pletely across the face of the boiler not in use, when steaming with one boiler in a fireroom. This will be held tightly over all openings by the air pressure, and prevent excessive loss through them.

The blower housings are designed to throw the air out over the boiler casings, and so warm it before it reaches the registers, but when operating in tropical climates this makes an excessive heat on the floor plates. To overcome this and preserve the health of the personnel it is usual to rig a wingless ventilator for each blower, which will bring a little of the fresh air down to the men on watch. In the *Jarvis* the temperature of the working platform was reduced from about 120–130 degrees to less than 110 degrees in this way. For port use a large winged ventilator will usually furnish sufficient air and make the fireroom livable, except in calm weather, when a blower must be used, not only for the boilers, but to prevent the men on watch being overcome by the heat.

The brick cones must be kept smooth and round, otherwise the projections will catch the spray and make the boiler smoke. The boiler bricking must be carefully examined after every run, and any tendency on the part of the brickwork to loosen up or fall down corrected at once by repairing the wall, using new bricks and high-temperature cement when necessary, or setting up on the fastenings of the old ones, the whole being thoroughly washed over with a preparation of silicate of soda and carborundum sand. This should be mixed as follows:

Carborundum fire sand, 65 per cent.
Ground fire clay, 20 per cent.
Silicate Soda, 52 per cent. Baumé 15 per cent.

If this solution shows any tendency to burn off the back wall, leaving the brick exposed in spots, or even taking part of the brick off with it, your burner adjustment or your oil pressure is wrong. If Bureau burners are installed you may be able to draw them out slightly, but in either case you must reduce your oil pressure so that the maximum combustion will

come in the center of your furnace. This can usually be judged by the soot deposit on the tubes, which is thickest near the center of the combustion area. Tubes should, of course, be blown as soon as the boilers are cool, and before putting on the smoke pipe covers, if possible.

In most of our small-tube boiler installations, about forty minutes in warm weather and one hour in cold weather is sufficient time for raising steam without danger to the boiler, but, of course, it can be raised in ten minutes in an emergency.

Each and every installation has its own peculiarities, and the engineer officer will be well repaid who puts time and study into the proper operation of this most vital part of his plant, and the fireroom crew of a modern oil-burning ship requires the same careful drilling and attention that the gun crews get. It is a great advantage to keep the watches together, and to stimulate, if possible, by competition, the interest to excel in economical operation, both underway and at anchor.

NOTES ON PUMPS.

By Lieutenant S. M. Robinson, U. S. N., Member.

The following notes have been compiled from information obtained from officers in the service:

No attempt has been made to describe the various pumps in the service, as it is the intention to use these notes in connection with the actual operation of pumps on board ship where drawings and descriptions of the pumps will be available.

Practically all of the ships in the Navy are fitted with main air pumps of the bucket type, main circulating pumps of the centrifugal type driven by a vertical, reciprocating engine, and service pumps of the plunger type. The *Salem* has a Blake dry-air pump driven by a vertical reciprocating engine and a Worthington hot-well pump driven by a turbine. The *North Dakota* has a Weir monotype wet-air pump; originally she also had a Weir rotative dry-air pump, but that has been removed as it did not prove satisfactory. The *Jupiter* has an Alberger dry-air pump driven by a vertical, reciprocating engine and a turbine-driven Alberger hot-well pump. The *Neptune* has a centrifugal circulating pump and a Westinghouse-Leblanc air pump on one shaft driven by a turbine, and a turbine-driven hot-well pump. The *Wyoming* has a Weir single-acting, dual air pump with wet and dry cylinders. The *Arkansas* and several of the later destroyers have turbine-driven circulating pumps, and practically all the new ships are being fitted with these. The new battleships are being fitted with turbine-driven feed pumps. The *California* is to be fitted with motor-driven main air, main circulating, dry-vacuum, hot-well and forced-lubrication pumps.

Pumps have been supplied to the Government by many different manufacturers, among whom are Dow, Davidson,

Blake, Worthington, Warren, Snow, Cameron, Deane and many others. There are, however, many general rules of care and operation that apply to all of them equally well, and these will be given first and followed by notes of points that are peculiar to various makes.

STARTING AND OPERATING.

The operation of both bucket and plunger pumps is very simple. First see that the proper suction and discharge valves are open, then open the drains, then open the exhaust, and then admit steam gradually. After the pump makes a few strokes close the drains and bring the pump up to the proper speed slowly. If cushioning valves are fitted, these should be kept closed at slow speeds and opened slightly as it becomes necessary to speed up the pump.

If the pump will not start, close the throttle and look for a closed valve on the discharge line; if none is found closed, then disconnect the valve gear, and, after cracking the throttle, work the valve by hand; if the pump still refuses to start, close the throttle and examine the main steam valve to see if it has over-ridden and stuck; if the main steam valve is all right, then the plunger is probably frozen, particularly if the pump has not been in service for some time.

After the pump has been started it may be found that it won't take a suction; this will be shown by the jerky operation of the pump. In this case, first look for a closed valve in the suction line. Or, if it is a feed pump, it may be that the valves have been leaking from the feed line and the pump is vapor-bound; in this case take a suction from a double bottom for a few strokes to cool the pump or turn a hose on it. Most feed pumps are below the feed tank and have a suction head, but some of the older ones have a suction lift, and these may get air-bound and require priming before they will take a suction. The same thing applies to other pumps having a suction lift; those pumping salt water can usually be readily primed from the sea.

If the pump races without any appreciable effect on the pressure gage, then either the plunger leaks or something is wrong with the valves in the water end. It may be that a valve is broken or merely that the valves do not seat properly. If the pump has been running all right and suddenly loses pressure on one stroke, it is safe to look for a broken valve at once. In some cases, with feed pumps running fast, the valve rims have been pushed off and the guides forced down into the water passages. In the case of air pumps, if the pump works with a jerky motion and the vacuum gage follows the motion of the pump, then something is wrong with the foot valves; if the vacuum drops and the pump races, the bucket valves are probably gone; if the pump holds a reduced vacuum and works somewhat jerkily, the trouble is with the head valves. If an air pump works steadily most of the time but gives an occasional short stroke, it will probably be found that the stuffing box of the auxiliary valve is too loose and the weight of the rocker arm is sufficient to move the valve and reverse the pump.

It should not be attempted to run air pumps too slow as they will probably stop; the reason is that practically all the work is done at the end of the stroke and, in slowing the pump down during its stroke, the throttle may be closed so much that the pump will not have sufficient steam to finish the stroke.

Grunting or groaning in the steam cylinder of a pump is usually due to the steam getting behind the piston rings and forcing them out against the cylinder; or to a scored cylinder; or to cut rings; or to cylinders being out of line. Grunting in the water cylinder is generally due to the packing being too tight. The cure for these troubles will be considered later on.

Pumps taking a suction from the sea (or other pumps with considerable suction head) sometimes pound on the water end; frequently this can be stopped by closing the suction off slightly, or, better still, by putting heavier springs on the suction valves.

If air chambers are not fitted on the suction side of pumps

the water end will usually hammer; a snifting valve on the suction side of the pump will usually relieve this.

Always keep a pump running on full stroke; the methods of insuring this will be taken up later on.

In the case of poorly designed pumps that stop continually an oiler will frequently attempt to use a jacking bar for starting the pump with steam turned on it; this is a dangerous practice and generally results in wrecking the pump sooner or later.

SECURING.

In securing bucket or plunger pumps first shut off the steam, then the exhaust and then open the drains. Close the suction and discharge valves and, after the cylinders are drained, close the drains. Drains should never be left open, as the air will cause cylinders to rust. In securing feed pumps they should be tested for defects in the following manner: Bring the pump up to 30 or 40 pounds above the boiler pressure, then close the discharge and shut steam off the pump; if the pressure holds, then the water end is tight; if the pressure falls, then the water end should be examined for the following defects:

(1) Leaky water piston;
(2) Leaky valves;
(3) Weak relief valve spring;
(4) Leaky joints;
(5) Leaky stuffing box;
(6) Defects in the water chamber which allow water to pass.

CARE AND UPKEEP.

All pumps should be moved daily to keep them from freezing. In order to make sure of carrying this out, a pointer should be fitted on the crosshead and seven marks made on a rod which the pointer will pass. Each of these marks should be labeled with one of the days of the week; it is then always easy to see if the pumps have been moved.

Be sure that the steam and water ends of pumps are kept in line. Pumps secured to a bulkhead are much more apt to get out of alignment than those with an independent base and setting, such as air pumps. Failure to keep proper alignment is one of the greatest sources of trouble with pumps on board ship and is one of the most serious defects. A pump may have been nicely aligned in the shop and then pulled out of line when it was bolted to the bulkhead, or, after it was secured, the ship may have changed her shape sufficiently to warp the bulkhead and cause the same result. If the pump is run in this condition it usually results in scoring the rod and cylinders and breaking followers and bolts. The alignment of pumps should be tested occasionally by removing piston and plunger and running a line through the cylinders; especially should this be done during the first year of commission of a new ship and also if a pump is giving trouble by scoring the rod or cylinder or breaking followers.

Pumps should be made to run with a full length of stroke. This means better steam economy and better all around working of the pump, and it invariably means that something is wrong, either with the design or adjustment of the pump, when it cannot be obtained. A short stroke means incomplete cushioning and causes shoulders to form in cylinders and valve chests with resultant breakage of rings and followers. The proper valve setting must be carefully determined to make a pump take full stroke. In this connection the following thumb-rule for setting valves has been tried and found to be satisfactory: Place pistons in the centers of the cylinders or on half stroke and the valves the same. Then with the top collar all the way down to the tappet, the valve should be open at the top ¼ inch and with the bottom collar all the way up to the tappet, the valve should be open ¼ inch at the bottom; with the collars at equal distances from the tappets the valve should be at the center. Another method of setting valves is to place pistons and valve on the center, as before; then move each collar from the tappet one-half the width of the steam

port. In the case of pumps where the tappet moves the full distance of the stroke, the distance from collar to tappet would be ½ (stroke minus steam-port opening). After the valves have been set trams should be made for the collars so they can be reset without removing the valve-chest covers. It is also advisable to drill holes in the tap screws of the collars and run a wire through them after they have been set; this will prevent inexperienced men from moving the collars, and it should not be necessary to continually keep moving the collars while the pump is running; if this is necessary, something is wrong with the pump and it should be dismantled and examined. Of course, some pumps are not properly designed and, when adjusted to give full stroke at low speeds, will override and pound at full speed. The only remedy for this is to have two separate valve adjustments and two sets of cams so that the valves can be set quickly. If a pump runs on short stroke for any length of time difficulty may be found later on in making the pump take a full stroke; it will usually be found that a shoulder has been worn in the valve seat and this will have to be removed before full stroke can be obtained.

Valve gears should be frequently removed and cleaned with kerosene.

Valves should always be kept scraped true and the packing rings of piston valves kept free. It is highly important that the steam valves should be kept tight; if the control valve leaks, the main steam valve will become steam-bound and the pump will stop. In facing off the pilot-valve seat of a Blake pump care should be taken not to face off so much that the valve will not be able to seat itself. With the new Davidson valve gear it is very important to keep the tapered pin in the small exhaust valve set up hard enough to insure a slight friction on the valve chest, otherwise the pin will get loose and the valve will float back and forth on the small play allowed it, and the pump will run on a short stroke and frequently stop; this pin is intended to take up wear and will have to be adjusted occasionally. With the old Davidson gear of

the cam and pin type, the best results were obtained by case-hardening the cam and fitting the pin with a roller and keeping plenty of spares on hand. Main steam valves are frequently put together without lock nuts; in this case they usually fall to pieces after a short time; lock nuts should be fitted and set screws put in to hold the lock nuts.

When the pins or holes in valve levers wear excessively this will seriously affect the valve motion; if this wear occurs rapidly it is best to bush the holes with a tool-steel bushing.

Examine the valves in the water end frequently. When overhauling them, be sure that the springs are in good condition and set up to the proper tension and secured by split pins. See that the valves are true; try them for this with a straight-edge; if this shows the valves to be dished or warped do not turn them, but put in new valves if they are made of metal, as metal valves are apt to break when they have been dished; in the case of rubber valves, their life can sometimes be prolonged by trimming and turning them. Keep the valves clean; a light mineral oil makes a good cleaner and a lye or soda solution is good for removing caked or gummed oil from valves. Be sure that the valves are set to have a lift which will give a circumferential opening equal to the clear opening through the seat; this will never be more than ¼ of the diameter of the opening. In renewing valves the new ones may be of slightly different thickness from the old, thus giving the valve too much or too little lift; too little will usually cause the valves to squeal. In Blake pumps of the valve-pot water-cylinder type, it is very easy to make a mistake in assembling the valves. If the nut on the discharge-valve spring guard is set up before the valve-plate rod nut, then the collar on the valve-plate rod will not bear on the suction-valve spring guard, and the suction-valve plate will be free, and the pump will simply churn water back and forth in the suction chambers. Be sure that the valves and seats are smooth so as to give a perfect bearing surface. Some pumps have the valve seats secured only by making them with a driv-

ing taper fit, and these occasionally work loose; the remedy is to peen over the edge of the metal slightly. Sometimes the valves are placed with several in one plate and the joint between valve plate and pump cylinder is made with a ground joint; it is very difficult to make this joint tight, and if it is not, water will leak under the joint and score it; this can be reseated in a ship's valve reseating machine; a copper gasket for this joint seems to give better results. In pumps that have the valve seats screwed into the pump diaphragm they should always be screwed in with white lead, otherwise it will be almost impossible to get them out. In Davidson pumps the discharge valve seats in the water end are sometimes secured to the pump diaphragm by shoulders on the valve stems which are screwed into the suction seats and these seats have small flanges under which gaskets are fitted. The manufacturer supplies a rubber gasket, but this will very shortly be squeezed out and the seats will leak and hammer. Hard sheet packing will give better satisfaction than rubber, and lead better still if the water to be handled is cold. If sufficient time can be spared for it to set, red lead makes a satisfactory joint. In some cases this flange has given so much trouble that it has been necessary to fit new seats with a ground joint. In air pumps the stems of the valves are sometimes too loose a fit in the seat; this results in allowing the stems to work, and this breaks the wire securing the stems together and then all of the stems back out. This can be remedied by refitting the stems. The most trouble with the water-end valves usually is in the main feed pumps; frequently the stems of these valves will break so often as to become a serious matter. A good remedy for this has been found, and consists in cutting the stem off the valve and turning a groove in the top of the valve for the spring to set in; the stem is pinned to the guard at sufficient height to allow proper opening of the valve, and it then acts merely as a guide for the spring and a limiting device for the valve. It will be necessary to fit a new stem, as the old one will not be large enough to make a close fit in the guard.

In some of the older vessels the air pumps have soft-rubber valves; these get soft and stick in a short time and also collect oil rapidly; they should be overhauled and cleaned after every run and renewed frequently. A strong soda solution is best for cleaning these valves.

The rubber valves in use at the present time generally wear the hole in them too large before they wear out or get soft; by bushing this hole with lignum vitae (boiled in linseed oil) the life of the valve can be very materially increased.

Sometimes the rubber valves in auxiliary air and circulating pumps and fire and bilge pumps disintegrate rapidly, and it has been found that thin bronze plates, similar to those used in air pumps, will give good results in such cases. These valves are usually fairly noisy, however.

Some of the cast valves are quite heavy and give trouble due to heavy hammering; the substitution of light bronze discs will usually relieve this trouble also.

Water plungers should be kept tightly packed but not tight enough to make them groan; in putting in flax or Tuck's packing it is best to soak it before putting it in, but if there is not sufficient time for this, the packing must be fitted loose enough to allow for expansion. Water cylinders should be examined occasionally to see if they are scored and to see if the packing or rings are in good condition. Pump rods are sometimes screwed into the plunger and secured by a lock nut with a flat plate secured on top of the plunger to prevent the nut from backing off. This will still allow the nut and plunger to turn together and the plunger may back off and carry away. This can be prevented by putting a set screw in the nut. A copper washer under the nut will help secure the nut from turning and getting loose on the rod.

The troubles in the steam cylinder mostly come from the rings. With split rings the steam may get in behind the rings and force them out against the cylinder, causing the pump to groan and cutting the cylinder and rings. The remedy is to fit locking bolts which will allow the rings to open

out only to the diameter of the cylinder; it will be necessary to readjust this ring as wear occurs. In aggravated cases relief has been obtained by turning a groove about 1/32 inch deep and 3/8 inch wide about the middle of the ring and drilling 1/8-inch holes right through the ring; these holes will relieve the pressure of the steam. When split rings cannot be made to work, resort is usually had to a solid ring, but this is not very satisfactory as this will leak as soon as it wears a little. Diagonally-cut split rings seem to give better satisfaction than over-lapping rings; the latter generally break at the corner after being in service a short time, although this trouble can be reduced by rounding the corners of the shoulders. Another source of trouble is the breaking of followers and bolts; this may be due to bad alignment, but it is also frequently due to a weak follower; where this trouble is general in any one set of pumps on board ship, it is pretty safe to assume that the followers are too weak and a new follower should be tried on one of the pumps to see if it stops the trouble. This condition can sometimes be improved by fitting the piston with through bolts. Pistons frequently work loose on the rods; this is generally due to poor shop work; the rod will be so fitted that the shoulder brings up against the piston without giving a proper bearing surface for the taper part of the rod; if the rod is properly fitted the trouble will usually stop. If a jamb nut is not fitted, that should be tried also.

Great care should be exercised in setting up on glands; if the two sides are set up unequally the gland becomes tilted and scores the rod and sometimes the gland itself is broken. If a gland is so located that one of the studs is not readily accessible it is best to fit a gland nut so that it will be impossible to tilt it. In the case of outside packed pumps, where the gland is so important, it has been found best to fit a piece of pipe on each stud after getting the gland properly adjusted, in order to prevent inexperienced men from changing it. If a gland is set up too hard it will score the rod; if a gland continues to leak after it has been given a few turns on the nuts

the best thing to do is to let it alone till the pump can be shut down, and then renew the packing. When excessive corrosion takes place on the salt-water end of the piston rods, this can sometimes be reduced by fitting a packing ring in the stuffing box and using a grease cup on it. The use of monel metal for the rods also eliminates this trouble.

Trouble has occasionally been experienced with the relief line from the feed pumps; if this is piped back to the suction line close to the pump, in some cases when the relief valve lifts, it will blow the water away from the suction and cause the pump to become vapor-bound; this trouble can be stopped by taking the relief line back to the feed tank. In the case of an air or vapor-bound pump it has been found that a pet cock for getting rid of the air will be more effective if placed at the top of the air chamber on the discharge side of the pump rather than on the valve-chest bonnet; this is, of course, only true when the air chamber is near the pump. Air chambers should be fitted on both suction and discharge sides of pumps and pet cocks should be placed on both at the highest point to admit air to take the place of that which has been absorbed by the water. Air chambers should be located as nearly as possible with the opening opposite to the direction of flow of the water.

When feed lines are fitted with grease extractors a pipe connection should be fitted on the discharge side of the grease extractor and led to a gage at the feed pump. This will immediately show if there is anything wrong with the grease extractor.

Feed-pump regulators are sometimes connected direct to the water end of the pump, and usually this causes the regulating diaphragm to cut; the trouble can be remedied by piping from the discharge side of the feed heater to the regulator and fitting an air chamber on the regulator.

LUBRICATION.

A very small amount of cylinder oil should be used on the steam ends of the rods and all outside moving parts should be

lubricated with mineral oil. No oil should be used in steam or water cylinders or valve chest.

VALVES.

The present specifications for reciprocating pumps for the Navy calls for metallic valves of either manganese or phosphor bronze for the feed pumps and air pumps and hard or medium-hard rubber valves for all other pumps. The valves for the air pumps are usually rolled and those for the feed pumps cast. The rubber valves are made of vulcanized rubber containing not less than 30 per cent. Para rubber for the hard and 35 per cent. for the medium-hard valves.

PACKING.

The present pump specifications call for metallic packing for the piston rod and valve stem packing on the steam ends of all pumps.

For the water end either flax or Tuck's packing or a combination of alternate layers of the two seems best for the rod and also for the plungers of outside-packed pumps. For water plungers flax or Tuck's seems to be best for all pumps except feed pumps, where white metal rings are generally used. There is a great divergence of opinion as to the best packing for feed-pump plungers, many officers liking Tuck's better than the white metal and others preferring the white metal. The latter seems to be gaining in popularity, however. These rings can be carried in the form of cast cylinders and a ring cut off when desired. If solid rings are used instead of split ones, their tightness can be improved by cutting water grooves about ¼ inch apart, ⅛ inch wide and 1/16 inch deep; these rings should be fitted tight enough so that they will just drop through slowly of their own weight. If split rings are used they should be fitted with springs to hold them out and with a lock piece to prevent too great expansion. A satisfactory mixture for white metal consists of two parts tin and one part lead. These rings as fitted are sometimes very

narrow, and this usually causes them to cut the cylinder; if wider rings are fitted the trouble generally stops.

For steam pistons, diagonally-split cast-iron rings with locking piece to limit the expansion seem to give the best satisfaction. Methods of preventing steam from getting behind these have already been considered.

Piston valves are generally fitted with rings similarly to the steam piston, but these have sometimes given trouble, which has been eliminated by carefully fitting a solid piston valve; this will not remain satisfactory if the valve wears much, however.

For all cold-water pumps a sheet packing of rubber with cloth insertion is suitable for the water end. For the steam ends of all pumps and water ends of hot-water pumps compressed-fiber packing is used. When trouble is experienced with gaskets blowing out the trouble can usually be eliminated by turning a shoulder on the head and cylinder so as to make a male and female joint; this shoulder will prevent the gasket from blowing out. On the water cylinder the head with the stuffing box gives the most trouble by blowing gaskets, on account of an alternate increase and decrease of pressure on the gasket due to the motion of the rod; this will be aggravated if the stuffing box is unduly tight. A very reliable joint for this end can be made by fitting the shoulder as above described and then using a copper gasket cut out in one piece and put on by disconnecting the rod at the crosshead.

CIRCULATING PUMPS.

Circulating-pump engines are usually compound, direct-acting engines; the new battleships have forced lubrication for these engines. On many of the smaller ships the engines are single-cylinder and these are usually fitted with a flywheel; these sometimes give trouble by vibration; this can be remedied by putting a balance weight on the flywheel. It is usual to start a circulating-pump engine from a certain position and a jacking bar is provided for jacking them into position; oc-

casionally some one forgets to remove this bar before starting or attempts to jack the engine over with steam turned on, and this generally results in a serious accident.

Before starting the lubrication should be turned on, the engine jacked into position, the drains opened, the exhaust opened and then steam turned on slowly. Sometimes it will be necessary to give a quick opening to the throttle to get the engine started, but care must be exercised if this is done; make sure that the cylinders are well drained and warmed up. After the engine starts, open the overboard discharge and the main injection.

The speed of a circulating pump should be regulated according to the amount of circulating water needed. As a rule the overboard discharge should be between 10 degrees and 20 degrees F. below the temperature corresponding to the vacuum.

When securing, close the throttle, then the exhaust and then open the cylinder drains and shut off the lubrication. Close the overboard discharge and main injection valves. After the cylinders are drained close the drains; if the steam or exhaust valves are leaking it may be necessary to keep the drains open.

These engines run at a high speed and must be kept carefully adjusted. The cylinders should be opened frequently and wiped out with kerosene. The bearings should be adjusted at the first sign of a knock. After making adjustments the engine should always be run for a short time to see if there is any tendency for the bearings to heat.

These engines should be jacked over daily.

The principal troubles in the water end are due to corrosion of the pump shaft and the runner getting loose on the shaft. The keys on the runners should be secured by set screws. Zincs fitted in the pump chamber will reduce the corrosion.

The lignum vitae bushings for the bearings sometimes wear rapidly, and this puts the pump out of alignment; this should be checked frequently and a new bushing fitted when neces-

sary. This bushing should be well soaked before fitting it, otherwise it will bind when it swells and the shaft will squeal. Forty-eight hours is about the minimum time that can be allowed for soaking lignum vitae.

The pump casing should be removed at least once a year and the pump examined; dirt and marine growth will collect in the pump chamber and must be cleaned out. Every time the ship docks, the pump chamber, suction piping and as much of the discharge piping as possible should be cleaned as this growth very materially reduces the efficiency of the pump.

Sometimes a circulating pump will have an unbalanced end-thrust so that the runner end clearance will not be equally divided and the crank web will rub against the bearing on one side. The best way to overcome this difficulty is to fit a thrust bearing on the end of the shaft. A simple device for this consists of a small iron bracket bolted to the framing and carrying a brass plate which bears against the end of the shaft. The brass plate is held against the end of the shaft by a set screw in the bracket and is prevented from revolving by a stop. Lubrication can be supplied through an oil hole to the center of the brass plate and distributed over its surface by a groove cut across a diameter of the shaft end.

Trouble has been caused by the zinc plates in the condenser dropping down and getting into the pump casing, causing it to rattle badly and sometimes breaking a pump runner. If brass washers are fitted on the studs holding the zincs or through bolts with washers fitted instead of the studs, this will usually eliminate the trouble; also the zincs should be renewed frequently, and this will reduce the probability of such an accident.

SPECIAL TYPES OF PUMPS.

U. S. S. Wyoming—Main Air Pumps.

These are Weir dual air pumps. The main point requiring special attention is a spring-loaded valve on the discharge pipe from dry to wet cylinder. This valve is adjusted to maintain

a vacuum of about 20 inches in dry cylinder when the condenser has 28 inches vacuum, and this difference of 8 inches in vacuum is sufficient to overcome the friction in the cooler and to pass the water into the suction. This valve and spring must be kept in good condition to obtain satisfactory results.

U. S. S. Jupiter—Dry Vacuum Pumps.

These are Alberger, reciprocating-engine-driven pumps. To start up, first see the circulating water turned on; next see that the exhaust valve is open; next see that the atmospheric valve on the suction to the air cylinder is open and that the valve to the condenser is closed; next jack the engine over to starting position; next open cylinder drains; next turn on steam; after the engine has made a few revolutions, close the valve to the atmosphere and then slowly open the suction from the condenser; close cylinder drains. These pumps have an oil-relay governor operated by the vacuum.

The upkeep of the steam end of these pumps is the same as that of any reciprocating engine; they are slow speed and give very little trouble.

The discharge valves on the air cylinder should be frequently cleaned with kerosene as they become covered with a deposit of burned oil.

The air cylinder and piston should be examined and wiped out with kerosene once a month.

Hot-Well Pumps.

These are turbine-driven volute pumps made by Alberger.

These pumps should not be started up until there is sufficient water in the condenser to fill the suction pipes; a gage glass has been fitted on the condenser to show height of water in it. Before starting up make sure that the equalizing pipe from the top of the condenser to the suction side of the pump is open, otherwise the pump will probably be air-bound; also make sure that the ¼-inch water-seal pipe leading from the

discharge side of the pump to the lantern ring in the stuffing box on the water end is open, as otherwise this gland will heat very rapidly and score the shaft, and also the pump will suck in air; there is a pet cock on this pipe which will always show whether or not the pump is throwing water; when running, a little water should trickle out of the stuffing box, and if it does not, the packing is too tight and the gland should be eased up. After these valves are examined see that the oil reservoir is about half full, open the exhaust, see that the suction and discharge valves are closed and turn steam on slowly till throttle is open (the turbine is governor-controlled); then open the suction valve and then open the discharge valve; if this procedure is followed no trouble will be experienced in getting the pumps quickly free of air and to pumping water.

The packing in the water-end stuffing box should be renewed about once in two months to make sure that it does not get hard and score the shaft.

The thrust bearing should be examined about once a month.

U. S. S. Salem.—Dry Vacuum Pump.

This is of similar type to that on the *Jupiter* but was supplied by the Blake Company. The operation of the two pumps is similar, but the dry-air pump connection to the *Salem* is low enough so that occasionally the water level in the condenser gets up to it; this will be shown by a change in the running noise of the pump and by the discharge of water from the drain valve on the bottom of the valve chest (this valve should always be kept open); when this happens, the suction valve should be closed at once, as water in a rotative dry vacuum pump will probably wreck the air end. Then open the drain valves on the air cylinder and it will free itself of water. Then, as soon as the water in the condenser has been removed, the suction valve can be opened slowly and the drain valves closed.

The springs of the air cut-off valves should be examined and renewed frequently. These valves as originally installed

gave trouble due to flapping on the valve face, but this was remedied by putting in guide rods for them. These valves can be adjusted while the pump is running. The point of best running should be determined by trial and marked on the adjustment-gage strip.

Hot-Well Pumps.

These are Curtis turbine-driven pumps of Worthington manufacture.

These are of a similar type of pump to those on the *Jupiter* and the operation is similar to those pumps. These pumps are governor-controlled, but are also provided with hand-operated nozzles which increase the turbine efficiency at the low speeds.

The greatest source of danger is in the water seal; if the pipe gets clogged it will have to be disconnected and blown out; if the gland is not properly adjusted the seal water may be cut off from the seal; this can be prevented by putting a set screw under the edge of the gland and screwed into the main casting.

Care should be taken to keep the roller bearings well oiled and the collars adjusted to give the turbine rotor the proper clearance; this can be measured when the turbine is stopped by removing a cover plate and using a graduated steel wedge.

North Dakota.—Main Air Pumps.

These are Weir monotype. The care and operation of them is practically the same as for other bucket air pumps. They only take steam on the up stroke. When first installed they hammered somewhat at the end of the stroke, due to the fact that the steam pressure on opposite sides of the piston was equalized before exhaust took place allowing the vacuum to pull the plunger down with a slap. This was remedied by fitting a pipe at the top of the air cylinder with both ends of the pipe leading into the cylinder, one end being above the other enough so that when the bucket reaches the end of its

stroke it passes over the lower end of the pipe and thus cross-connects the two ends of the cylinder and equalizes the pressure.

U. S. S. Neptune.—Main Air and Circulating Pumps.

These are horizontal geared-turbine pumps of the Westinghouse-Leblanc type, the air and circulating pump being run by the same turbine with a 7 : 1 reduction, the turbine running at 3,500 r.p.m. The glands of the circulating pump are supplied from the discharge side of the pump and must be kept clear. The air pump has a reservoir for fresh water in its base and a cooling coil for this water which is cooled by the main injection. A valve in the fresh-water line regulates the amount of water to the pump. The air-pump glands are supplied with fresh water from the main feed line and these glands must be kept open and supplied as is the case with all pumps of this type.

Hot-Well Pumps.

These are direct turbine-driven pumps running at about 2,500 r.p.m. The operation and care of this pump is similar to that of the hot-well pumps on the *Jupiter* and *Salem*.

INTERESTING ACCIDENTS AND REPAIRS ON BOARD SHIP.

The main circulating-pump casing of the *Iowa* leaked; it was discovered that the metal where the chaplet rods had been in the mold was spongy; this was tapped out and plugged.

The main circulating pump of the *North Dakota* rattled badly and for a long time no reason could be found for it. The casing of the pump had a very sharp edge at the junction of the discharge pipe and the end of the spiral; this was smoothed up and the rattling ceased.

The water pistons of the main feed pumps of the *Vermont* originally consisted of a solid brass ring held in place by a follower plate; this piston was unsatisfactory and kept the cylin-

der scored all the time and was not tight; the brass ring was replaced by a sectional-cast white-metal ring with springs behind it, and this proved satisfactory. These rings were 6 inches wide.

On the same ship, while cruising at a long distance from a navy yard, two of the steam chests of the main feed pumps broke off short at the neck of the flange and on the end where the bell crank for working the valve gear was attached. In breaking, half of the flange remained attached to the steam chest and the other half to the bonnet. Repairs were made by the ship's force. The broken pieces of the flanges were put in place and an iron band fitted and shrunk around the flange to hold it in place; then long stud bolts were fitted from end to end of the flanges to hold the broken parts of the steam chest together. A sheet-iron casing was then fitted around the flange to about half the length of the steam chest, leaving a space of about two inches between the external side of the steam chest and the sheet-iron casing. This space was utilized for making a rust joint made of a mixture of cast-iron borings, sulphur, salammoniac and water. After setting for 24 hours the joint was solid and held when placed in service. This method of repair has been used on other ships in a number of cases for both valve chests and cylinders. When possible to do so, it is best to caulk copper wire into the crack on the inside of the cylinder. A good mixture for this job consists of 1 pound iron borings, 2 ounces of salammoniac powder, 1 ounce of flowers of sulphur; mix the whole dry; use this mixture with 20 pounds of iron borings, mixing with water to the consistency of mortar.

On the *San Diego* one of the pumps always had a heavy thump on the bottom stroke. The trouble was finally located in the liner; it did not fit at the bottom of the cylinder and there was a ¼-inch hole through the cylinder wall connecting to the suction port. The hole was plugged and the thumping stopped.

On the *Resolute* the spiders that held the packing in the

water plunger out against the walls of the cylinder carried away. The ship had no spares and no machine shop. Pieces of wood were cut to fill in the space between the piston head and packing; when the wood swelled it held the Tuck's packing out against the cylinder and gave satisfactory working.

Some of the older ships, fitted originally with wick feed lubrication, have been changed to forced feed. The result of the change in some cases was to make it impossible for the throttle man to see whether or not his auxiliaries were running. To overcome this tell-tale lights were fitted up by putting a make-and-break connection on the beam of the air pump and on the indicator connection of the circulating pump.

On the *San Diego* trouble was experienced with the auxiliary air and circulating pump; it stopped frequently. It was found that in putting a stud into the valve chest the button had been punched out of the hole; when steam was turned on this button would blow onto the port. It took some time to discover this, as the button would fall away out of sight as soon as steam was shut off.

On the *San Diego* the hand reversing gear of the main engines did not work satisfactorily, and finally the hand pump was changed as follows: The oil reservoir was removed, the packing of the oil piston was changed to Tuck's, and a connection brought from the discharge side of the main feed pump to the hand-pump plunger cylinder. An exhaust connection was made from the main feed tank to the nozzle on the pump formerly used as a reservoir connection. Thus the oil pressure produced by the hand pump was replaced by the feed-line pressure. This gave good satisfaction.

In the water end of the *Connecticut's* feed pumps the valve stems broke frequently. They were screwed into the suction seats and had a collar for securing the discharge seats. The pump diaphragm between suction and discharge valves was weak and would pant, bringing a heavy strain on the stems and breaking them. Screw stays were fitted between the seats and this stopped the trouble.

On the same pumps the flanges on the cylinder bonnets, chests, and also on the distance pieces between the water legs were not wide enough and gaskets would blow out continually. Heavier bonnets were made and as many stud bolts as possible fitted and this relieved the trouble.

These pumps were outside-packed and the stuffing boxes were too shallow; this resulted in broken gland studs due to attempts to set up too hard on them and also in scored plungers due to too much pressure on the packing. The trouble could not be eliminated, but was reduced by using a special packing of vulcanized rubber made in rings to fit the stuffing boxes exactly.

NOTES.

RECENT PROGRESS IN MILITARY AERONAUTICS.*

By Lieut.-Col. Samuel Reber,

In charge of the Aviation Section, Signal Corps, U. S. A., Member of the Institute.

The first decade of the twentieth century has added to the mechanical engines of war two types of apparatus, one heavier than and the other lighter than air. While the rigors of warfare will be apparently increased by their use, they will prove potent factors in the disappearance of war from the forum of the world as the ultimate means of arbitrament for international controversies. The application of mechanical flight to warfare is but an added proof of the maxim of the German strategist who said, "War is the only science that lays under tribute all other sciences." Any progress in mechanical arts that tends to shorten the duration of a war, even by increasing the destructive potentiality of its implements, is a real gain to humanity, and the discoverer of a process or the inventor of an engine which increases its horrors and destruction is a real benefactor to mankind, since the increased loss of life and property will certainly cause civilized nations to hesitate and seek other less drastic methods for the settlement of international differences before final appeal is made to the arbitrament of arms. The art of mechanical flight has extended the domain of warfare into three dimensional spaces, has created new means of observation, communication and attack, whose potentialities are now but partially realized by the great military powers, whose existence has brought into being a new arm of the service, and whose utilization has produced great changes in the military and naval policies of the great powers.

As sea power has been one of the dominating factors controlling the policies of great powers in the past, air power will be so in the future. The moral effect of the presence or absence of an efficient air force will be a deciding factor in hostile operations, and the commander who is without an efficient air fleet will be at a great disadvantage from the outset of operations, and will have the celerity and certainty of his movements greatly hampered by the feeling that his plans and disposition of his forces are known to his opponent through aërial reconnaissance. To achieve success without an aërial force, he must greatly outnumber his opponent. Information of the location and movements of the enemy are absolutely essential to any commander. Such information is usually obtained by the active use of cavalry in contact with the enemy or by secret service within his lines. An efficient air force will not only supplement but anticipate the reconnaissance work of the cavalry, and can penetrate miles beyond the line of contact, provided it is not opposed by similar and better aircraft of the enemy.

The two best examples as to what might have happened had an army been supplied with a fourth arm were in the recent Russian-Japanese

*Presented at a joint meeting of the Mechanical and Engineering Section and the Aëro Club of Pennsylvania, held March 26, 1914.

war. Had the Russian commander had an aërial reconnaissance force at his disposal it is more than probable that the results of the two decisive battles of Liao Yan and Mukden would have been entirely different. Had Kuropatkin checked the Japanese advance on the Tai-tse-ho in September, 1904, the outcome of the war would have been much more favorable to Russia. On August 25, 1904, the Japanese attacked the Russian position in front of Liao Yan, and on the 27th the Russians, after an obstinate defence, retired to a second position, which was attacked without results until the 31st, when Oyama decided to send the first army under Kuroki to turn the left wing of the Russian position. On the 31st of August and 1st of September Kuroki's army moved to the northeast and separated itself completely from the main position by a distance of ten miles. This movement was absolutely unknown to the Russian commander, and, by its attack on the 2d of September, so relieved the situation along the main front that the Japanese advance, which had been completely arrested by the Russians, was renewed and Kuropatkin compelled to retire along the railroad to prevent his army being cut off from its line of communication. Kuropatkin's information had led him to believe that he was outnumbered at the beginning of the engagement by the Japanese, while, in fact, he had a decided superiority in men and guns. It must be evident that a few hours of aërial reconnaissance would have disclosed the real state of affairs and indicated the isolated position of Kuroki on September 1, and given Kuropatkin a splendid opportunity to capture or defeat with ease the entire first army. Again, in front of Mukden, Kuropatkin was unaware of the arrival of the third army, and even of the existence of the fifth army, nor did he have any information as to the extensive flanking movement of the third army from the 21st of February to the 1st of March, which movement ultimately compelled his withdrawal from his fortified lines. One single speed scout could have started from Kuropatkin's headquarters, circled the country over which the flanking movement was made, and returned and reported the movement within an hour and a half.

The functions of aircraft in war are fourfold: reconnaissance, both strategical and tactical; the prevention of similar reconnaissance on the part of the enemy; for communicating purposes, and for destructive action against the enemy.

The strategical reconnaissance by aircraft will cover all operations for the acquirement of information as to the location and movements of the enemy, and also of the terrain. When the reconnaissance becomes tactical in nature, as for example during an engagement, it will ascertain the location and disposition of the opposing troops, especially the reserves, and artillery positions, and inform the artillery commander of the effect of the fire of his guns. Whether reconnaissance will be made by dirigibles or aëroplanes will depend upon the available aëronautical material at the disposition of the commander general, the weather conditions and the special object of the particular reconnaissance. For long-distance work, and especially for over-sea operations, the dirigible possesses certain desirable features, especially as part of the work is to be done at night. For the prevention of reconnaissance on the part of the enemy there will be required types of fighting machines which can carry light automatic guns and possess a greater climbing ability than those of the opposing force.

For communication purposes the aëroplane is especially suitable. The conditions of modern warfare—the large extent of territory over which forces operate, and the length of the line of contact in a modern battle—render it extremely essential for the commanding general to have the swiftest means of communication, which is best furnished by aëroplanes.

The use of the aëroplane and dirigible for destructive operations offers a large field in the attack by explosives from overhead of permanent

works, depots of supplies, dockyards, railroads, cities, and troops in position, but in the latter case the effect will be more moral than physical. A squadron of dirigibles or aëroplanes can compel a blockading fleet to remain a safe distance away from the shore, keep it constantly in motion, and cause greater coal consumption—a dominating factor in naval operations.

The possibilities of destructive aërial attack are well given in the following extract from a recent paper on aërial warfare by an officer of the French Army, Lieutenant Sensever:

"We shall see later on the possible utilization of the German aërial fleet for the destruction of works of art, arsenals, stations, or the direct struggle against enemies terrestrial or aërial. For the destruction of a city no problem of target practice arises; all shots count; the target is vast; no projectile is lost; no problem of defence is necessary for the aërostat, because in the night time it is invisible; no cannon can reach it; no aëroplane can pursue it; it is heard; but cannot be seen. Let us imagine on the evening of the declaration of war the ten Zeppelins, now in service, set forth with lights extinguished at an elevation of 1,000 meters towards Paris., The compass, the luminous signals of the stations and railroads, the lights of the cities are guides which indicate the direction to an aërostat more clearly, in fact, by night than by day. In the silence of the country, in the nocturnal quietude after the sounds of the city, the distant roar of the motors will vanish, leaving behind it incertitude. What is there to do? The Zeppelins steer towards Paris. In four hours, if there be no wind, they will cover the distance which separates the Rhine from the French capital. Now they are above Paris, whose innumerable lights mark out the quarters, the squares, the avenues; then it is very simple. When each one arrives at its appointed spot it accomplishes its task of destruction and death: to one, the stations where our soldiers will embark tomorrow; to another, the Ministry of War, the Elysée; to others, the forts or barracks at Versailles or Vincennes; to others, the Opera, the Louvre, Notre Dame, the Arc de Triomphe—all our glories and our forces! In a few minutes each Zeppelin has discharged its four or six bombs of 600 pounds each, in all about 50, or about 30,000 pounds of trinitrotoluol, which will level the beautiful quarters, which will kill millions of sleeping people, which will destroy gas mains, sowing conflagration and confusion in the terrorized city. During these few moments the Germans, from their nacelles, will contemplate their work—this spectacle that Nero never dreamed of. Then they will take their journey towards the east. At daybreak they are resting in their hangars along the Rhine, ready to start out again tomorrow."

The above is not an idle speculation, but could happen if the French are unprepared to meet the invasion of their frontiers through the air. England has this same menace threatening her across the North Sea.

Since the beginning of the twentieth century aëronautics have been used in three wars: the Russia-Japanese, the Bulgarian war and the Turko-Italian, and in minor expeditions of the French and Spaniards in Africa. The dirigible had not been sufficiently developed and the aëroplane was not in existence in the Russia-Japanese war, in which the Russians used captive balloons to advantage for observation purposes and for the direction of their artillery fire. The Japanese, though in possession of a balloon train, made practically no use of it during the war, though at Port Arthur there was an excellent opportunity for its use in locating the Russian fleet. Had the Japanese used a captive balloon the fleet could have been easily located and the siege expedited without the terrific loss imposed upon them by the capture of 203 Metre Hill, from which they obtained a view of the harbor and the ships. In the Turko-Italian war in Africa the Italians used both dirigibles and aëroplanes. These dirigibles, P-2 and P-3, only 4,100 meters in size, made in all 91

flights, and P–3, engaged in the battle of Zanzur, discovered and routed a force of Turko-Arabic cavalry by dropping bombs. These dirigibles, assisted by aëroplanes piloted by Italian officers, were of the greatest value for reconnaissance purposes, and, though frequently exposed to hostile fire, escaped without material damage.

In the Balkan war it may be said that practically no effective use was made of aëroplanes by any of the belligerents, as they were entirely unprepared for this class of service. In the first war the Turks had two in front of Kirk-Kilissé which were broken down and did not fly. They had some others in the rear which were not used until after the Bulgars reached Chataldja. The Bulgars had no trained aviators, and were compelled to hire foreigners, who rendered them practically no service. The advantage to the Bulgars of a single aëroplane is illustrated by that period of loss of contact, October 24 to 27, 1912, when the Bulgarian cavalry broke down and they had no knowledge of the Turkish position.

At Adrianople the Turks had no aëroplanes. The Bulgars used a number which were mostly manned by foreigners. General Ivanoff summed up their use as follows:

"The best that can be said for the aëroplane is that the flights showed what great service they are capable of if well manned."

The Servians, while having four aëroplanes, did not put them in service. At Chataldja a number of flights were made for the Turks by foreign aviators, and the Greeks made several reconnaissance flights in their operations.

The lesson from the Bulgarian war is that material cannot be obtained and military aviators trained after the opening of hostilities, and that foreign or civilian aviators cannot be depended upon for efficient service. Experience has shown that for military aviation the pilots as well as the observers must be trained soldiers.

The first really successful ascent of a dirigible was made in September of 1884 by the "La France," built by Captains Reynard and Krebs. This balloon was only 51 meters in length, with a cubic capacity of 1,864 meters, and driven by a 9-horsepower electric motor, giving it a speed of 14½ miles per hour. It was not until the beginning of the twentieth century that the French began the development of the present non-rigid type of balloon and added this to their military forces. The Germans, whose attention was drawn to the remarkable progress made by the French since the beginning of the century, began the development, under Count Zeppelin, of the rigid type of balloon. Experience has developed three separate types: the rigid, the semi-rigid, and the non-rigid, each possessing certain advantages and disadvantages for military purposes.

The rigid type is composed of a rigid frame of aluminum or wood in whose interior are a number of gas bags. The frame is covered by a light, permeable material, and the nacelles, carrying the motors, equipage and crew, are attached to the frame. The Zeppelins have a frame of aluminum, and the Schutte-Lanz and Speiss types of wood. The envelope of the semi-rigid is completely flexible, and its form is maintained by the pressure of the inclosed gas, while the weight of the nacelle and the equipment is distributed along the envelope by means of a keel or longitudinal girder, as in the Gross, Lebaudy and the Italian military dirigibles and those of the Forlanini type.

In the non-rigid the nacelle is directly suspended from the envelope, as in the Astra Torres, Parseval, Clement-Bayard, Zodiac and the Siemanns-Schukert.

The rigid type possesses the advantage of greater speed, more inherent stability, and, on account of the construction, the possibility of carrying armament not only in the nacelle but on the upper surface; is less liable to damage from hostile fire, owing to the fact that the gas is not contained in one but several separate bags. The rupture of one does not

cause the loss of the entire volume of gas. The rigid carries a larger crew and can remain longer in the air than the other types. The rigid type possesses the disadvantage of high cost, is more difficult to handle in starting and landing, requires a larger number of trained men in handling it in these operations, must be housed when not in motion, is more liable to damage or destruction in unfavorable weather, and more liable to accident, usually total destruction, in making bad landing, as evidenced by the accidents that have happened to the Zeppelins.

The semi-rigid offers a greater resistance, has less speed than the rigid but greater weight economy for equal volume, and is more easily handled and much less liable to damage in landing. It, as well as the non-rigid, can be anchored in the air when not in use, and is transportable, although the keel renders it less so than the non-rigid. The non-rigid has a slower speed in the air than the semi-rigid, is the easiest to handle, has the greatest economy of weight, and possesses the least liability to damage in landing, and is easily repaired and transported. The tendency is towards two types: first, the large, rigid, armed cruiser type, with a radius of action of 500 miles and an endurance in the air of thirty or forty hours. This type is assigned to permanent stations along the frontiers, with a zone of observation and defence. The second is a small, non-rigid type for over-sea operations. The Germans have developed this type of balloon more rapidly than the French and English, as it seems to be part of the German policy to offset England's supremacy on the seas by a German supremacy of the air. There are under construction in Germany dirigibles of this type up to 30,000 cubic meters capacity, with a lifting power of fifteen tons.

Since what may be called the real beginning of aviation in the year 1909 (Fig. 1) the records indicate a tremendous development in speed, distance, duration and altitude. The relative importance that the various great powers have attributed to aëronautics in war may be inferred from the following (Fig. 2), which shows the amounts of money expended by them during the past five years ended in August last.

Our own country was the first to realize the military importance of the aëroplane, and in 1907 the War Department issued specifications and proposals covering the construction of an aëroplane for the military service. In 1908, in the trials at Fort Myer, Va., due to an unfortunate accident, Lieutenant Selfridge was killed and Mr. Orville Wright injured, and activities were suspended until the following year, when the Wright Brothers demonstrated their machine, which was accepted by the Government. Practically nothing was done in this country until after July, 1910, when the first appropriation of $125,000 was available. A few machines were purchased and a few officers placed under instruction at the aviation school at College Park, Md. In 1911 a small aviation detachment was sent to the Texas border at the Camp of Concentration, and a number of reconnaissances made of the Rio Grande frontier. In 1912 they were used in the maneuvers in Connecticut and also for the observation of artillery fire at Fort Riley, Kan. In 1913 they were again sent to the concentration camp at Texas City, Texas, and the school detachment subsequently moved to the present establishment at San Diego, Cal., where the training has been continued and detachments sent to the Hawaiian and Philippine Islands. While the number of officers on this duty has been small and the equipment extremely limited, the results obtained by them, I think, are on a par with anything that has been achieved in the military establishments abroad.

Aviation has appealed to the *esprit* of the French nation in a stronger way than to that of any other nation. The French have led in the development of military aviation perhaps to offset the German superiority in dirigibles. Military aviation in the French Army was in 1912 made a separate arm of the service, it having been under the Engineer Corps.

It is directly under the War Ministry, and has a general officer at its head. The personnel is composed of officers and men recruited from all arms of the service, and when the new organization is completed the French Army will be in the possession of twenty-seven field detachments, five fortresses, and six coastal detachments, for which 450 machines and twenty-five dirigibles are altogether available. The service is divided into two branches—that of the dirigibles and that of the aëroplanes. The headquarters of the French aëronautical service is in Paris, and the whole of the military aëronautical establishments in France and Northern Africa are contained in three groups, with headquarters, respectively, at Versailles, Rheims and Lyons. The field detachments are assigned, as a rule, to the headquarters of the various army corps. The Central Supply

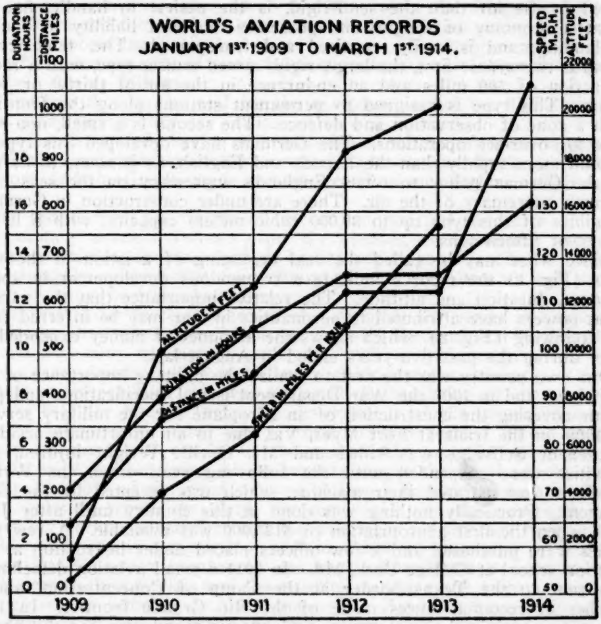


FIG. 1.

Depot and aëronautical laboratory is at Chalais Meuden. The personnel of the flying corps at present is 248 officers and 1,374 enlisted men.

The military aëronautical troops of Germany are under the command of a general of division, and the service is divided into that of dirigible balloons and that of aëroplanes, each under a separate head at Berlin. The total personnel is 82 officers and 1,458 non-commissioned officers and enlisted men. Germany is divided into three grand groups, the first facing towards Russia, the second towards France, Colmar, Metz, Strassburg and the Rhine, the third constituting the entire central hub near Berlin. These troops are divided into four battalions. The first battalion is located near Berlin and at Dresden, the second at Posen, Graudenz and Königsberg,

the third at Cologne, Hanover and Darmstadt, and the fourth at Strassburg, Metz and Friedberg. Bavaria will have a separate battalion. The present aëronautical project calls for an expenditure of twenty millions by the first of January, 1918.

In Great Britain the military aëronautical establishment consists of the Royal Flying Corps, which comprises the central flying school at Upavon, Salisbury Plain, the naval and military wings, the reserve, together with a separate establishment, the Royal Aircraft Factory, and the government aëronautical laboratory. Control of the military policy is under the Air

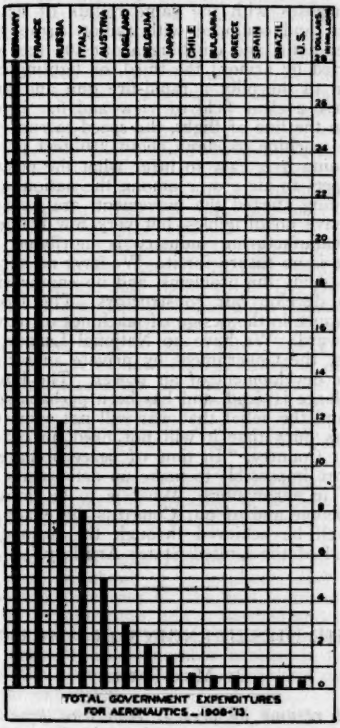


FIG. 2.

Committee, which is a sub-committee of the national defence committee. The military wing of the Royal Flying Corps is under the newly-created Department of Military Aëronautics in the War Office, while the naval wing is now managed by the air committee of the Admiralty. So far, no one has been appointed to supreme command of the entire flying corps, as was contemplated in the original scheme, although there is at present some agitation for the establishment of an air department, with a responsible head filling the functions of Minister of Aërial Defence.

There are five aëro squadrons in the military wing, three of them located at South Farnborough, one at Montrose and one at Nethersdon, and eight stations for the naval wing. There are available for service six dirigibles and one hundred and sixty-one aëroplanes. On the first of January of this year all the dirigibles were transferred from the military to the naval wing of the Royal Flying Corps.

Experience has developed three types of aëroplanes for military purposes: The first, the speed scout, for strategical reconnaissance, a one-seater, with speed up to 85 miles per hour and radius of action of 300 miles and a fast climber, about 700 feet per minute; the second for general reconnaissance purposes with the same radius of action, carrying both pilot and observer, and equipped with radiotelegraphy, slower in speed, about 70 miles per hour, and climbing about 500 feet per minute, and in some cases protected by armor; the third, or fighting craft, armored, and carries in addition to the pilot a rapid-fire gun and ammunition, and so arranged as to have a clear field of view and fire in either direction up to 30 degrees from the line of flight, the speed to run from 45 to 65 miles per hour, and the machine to climb about 350 feet per minute.

From the foregoing it is easy to infer what the position of the United States is with respect to air power among the nations of the world. We, who in the beginning started the movement, are now at the tail of the procession. We have no dirigibles, but very few trained men, and fewer machines. The manufacturing industry is moribund from the lack of business, and there is no future for it. We have no aërodynamical laboratories in which to study the problems, and no engineering courses, except one, in which to develop our constructors. The Government has not stimulated any advance in the design of machines or motors by competition for substantial reward. We have no National League, as in France and Germany, to assist the Government by private subscription and by public demand for the development of air power. The interest of our people in aëronautics at large is dead, and has been perhaps so lulled by a sense of false security and the belief that war will not come to such a vast and powerful nation as ours that it will not heed an oft-quoted maxim of the Father of Our Country, "In time of peace prepare for war." In no particular is it more impossible to make up deficiencies after the outbreak of hostilities than in aëronautics. What is to be done? To call to the attention of our people and of our Congress the exact state of affairs today. This can well be achieved through the membership of your institution.—"Journal of the United States Artillery."

THE DIESEL ENGINE IN AMERICA.

By Max Rotter.*

SYNOPSIS.—A résumé of present American practice as regards details of design, from a paper presented at the International Engineering Congress.

"Ignition by compression," so generally and superficially accepted as the fundamental principle underlying Dr. Diesel's invention, was, as a matter of fact, merely a natural corollary, and was not even claimed by him as his original conception, although typical of the engines that bear his name and successfully employed in them alone. Dr. Diesel's great aim was a closer realization of the theoretical efficiency of the Carnot cycle than attainable by steam engines or internal-combustion engines of the ordinary types. And although his aspirations were crowned with only

*Chief Engineer, Busch-Sulzer Bros. Diesel Engine Co., St. Louis, Mo.

partial success, it is scarcely to be doubted that they created a heat engine that has the highest efficiency that will ever be attained with fuel.

Carnot's law, as applying to heat engines, is well understood—the higher the initial temperature and the lower the terminal temperature, or the higher the initial compression pressure and the lower the terminal pressure, the higher will be the theoretical thermal efficiency. Dr. Diesel demonstrated that, although the theoretical and thermal efficiency increased slightly beyond this point, the highest mechanical efficiency was attained with a compression pressure of 30 atmospheres and a compression temperature of 500 degrees C. and, balancing these advantages against one another, that the highest actual efficiency and the greatest useful power development per unit of cylinder volume would be obtained by compressing the air to a pressure of between 30 and 40 atmospheres and a temperature between 500 and 600 degrees C.; and introducing the fuel in such manner as to obtain combustion at constant pressure with a maximum combustion temperature between 1,600 and 1,800 degrees C. On either side of these limits the actual efficiency of the engine diminished. It is evident that the stated compression temperature is "far in excess of that required" solely for the ignition of any carbonaceous substance that would ordinarily be used as fuel; also that, to avoid ignition during the progress of the compression, the fuel must not enter the cylinder until after the full compression has been attained.

In 1894 the first successful experimental engine operating upon these principles was completed by Dr. Diesel. In 1897 Adolphus Busch, of St. Louis, purchased outright the sole rights to all Dr. Diesel's existing and future United States patents and introduced the Diesel engine into this country under the sponsorship of the late Col. E. D. Meier.

The first Diesel engine built in the United States was completed in 1898. Since then these engines have been installed in municipal and commercial central power stations, in the most diversified industrial plants, and as stand-bys for water powers, in capacities up to 1,000 H.P. In Europe stationary Diesel engines have been installed in units up to 4,000 H.P., but the relatively lower prices of fuel in the United States tend to retard a similar development here.

Following in the wake of the Diesel, or high-compression constant-pressure oil engine, has come the so-called semi-Diesel, or low-compression oil engine. The distinguishing characteristics of the two types may be briefly stated as follows:

In the Diesel the entire cylinder volume of pure air is compressed to the maximum pressure of about 35 atmospheres and the corresponding temperature of about 500 degrees C., the fuel is introduced at or about the completion of this compression, and is gasified and ignited by the heat of compression; the combustion taking place without any material increase in pressure, but with a considerable increase in temperature, during 8 to 12 per cent. of the piston stroke, and continuing during a subsequent period of isothermal expansion.

In the semi-Diesel engine the entire cylinder volume of pure air is compressed to about half the compression pressure usual in the Diesel; a small portion of this air being contained in an auxiliary chamber that is in open communication with the interior of the cylinder and that has been heated to a high temperature (by external means prior to starting, or by the heat of the previous combustion while running), the temperature resulting from the mechanical compression being, therefore, considerably higher in this chamber than in the main portion of the cylinder. The fuel is introduced at or about the completion of this compression, either directly and entirely into this auxiliary chamber or through the cylinder and partially into the chamber, and is gasified and ignited by the heat of compression of the air in the chamber, the combustion taking place suddenly and with a greater increase in pressure than in the Diesel and being followed by a more rapid drop.

In engines of both types the maximum ignition pressure is about the same, but the combustion temperature at the commencement of the adiabatic expansion is higher in the Diesel. Figs. 1 and 2 show, respectively, typical indicator diagrams of a Diesel and a semi-Diesel engine.

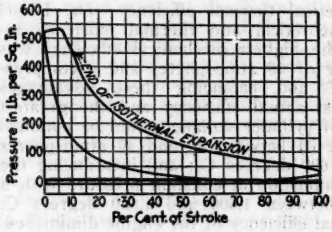


FIG. 1.—DIAGRAM FROM FOUR-STROKE CYCLE DIESEL ENGINE.

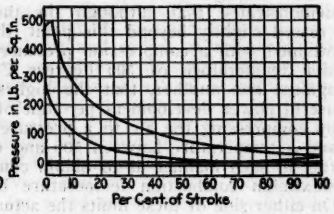


FIG. 2.—DIAGRAM FROM FOUR-STROKE CYCLE SEMI-DIESEL ENGINE.

The two-stroke cycle, as well as the four-stroke cycle of operation, is employed in Diesel engines. The lower efficiency of the two-stroke cycle is due primarily to the power consumed by the scavenging pump, practically none of which is recovered in the working cylinder. This is partly offset by the reduced mechanical losses due to lighter moving parts. The drop of pressure at the release, when the exhaust port of the two-stroke cycle engine is opened, is partially counter-balanced by the "bottom-loop" negative work of the four-stroke cycle.

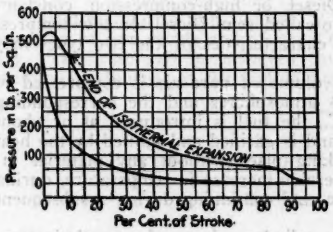


FIG. 3.—DIAGRAM FROM TWO-STROKE CYCLE DIESEL ENGINE.

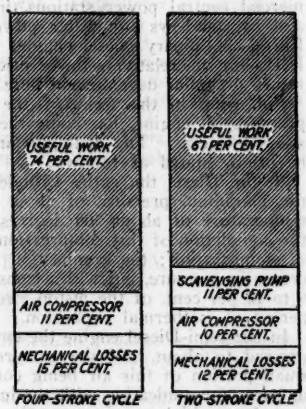

USEFUL WORK 74 PER CENT.
AIR COMPRESSOR 11 PER CENT.
MECHANICAL LOSSES 15 PER CENT.
FOUR-STROKE CYCLE
USEFUL WORK 67 PER CENT.
SCAVENGING PUMP 11 PER CENT.
AIR COMPRESSOR 10 PER CENT.
MECHANICAL LOSSES 12 PER CENT.
TWO-STROKE CYCLE

FIG. 4.—HEAT DISTRIBUTION IN TWO- AND FOUR-STROKE CYCLE ENGINES.

Fig. 3 shows a typical diagram of a two-stroke cycle Diesel engine and may be compared with Fig. 1, which is that of a four-stroke cycle engine. Fig. 4 illustrates the proportional subdivision of the indicated power developed by four-stroke cycle and two-stroke cycle Diesel engines.

The curves shown in Fig. 5 compare the fuel consumption of four- and two-stroke cycle Diesel engines built by the same American manufacturer. The older designs of two-stroke cycle engines, which were provided with cam-opera.ed scavenging and exhaust valves, were run at speeds considerably below those which are usual in the equivalent four-stroke cycle machines, because the camshafts of the former revolve at the same speed as their crankshafts, whereas those of the latter revolve at but half the speed. In later designs this limitation has been practically overcome by the substitution of scavenging and exhaust ports in the cylinder wall, controlled by the piston, in place of the heavy valves. This construction, although familiar in gas-engine practice, was adopted hesitatingly for Diesel engines, on account of their somewhat high exhaust temperatures, the supposed difficulty of constructing suitable cylinders, and the inefficient scavenging resulting from the fact that the scavenging ports had to be uncovered after the exhaust ports were open, and therefore re-covered before the exhaust ports were closed.

In Fig. 6 is illustrated the construction of a modern two-stroke cycle Diesel-engine cylinder provided with a recently patented arrangement of scavenging and exhaust ports. The exhaust ports are controlled by the

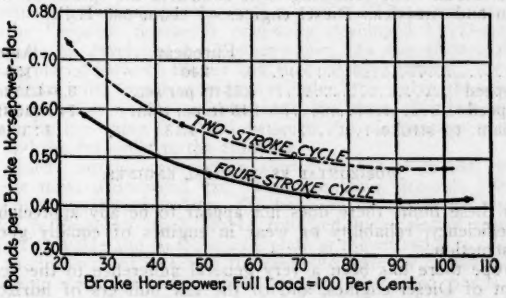


FIG. 5.—CONSUMPTION CURVES FROM SNOW ENGINE.

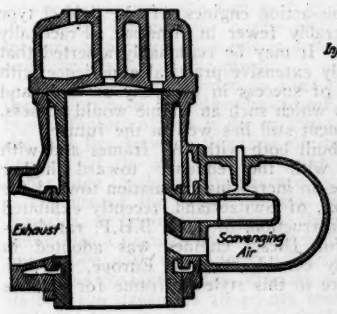


Fig. 6.

Fig. 7.

Fig. 8.

FIG. 6.—CYLINDER OF TWO-STROKE CYCLE ENGINE WITH AUXILIARY SCAVENGING PORTS. FIG. 7.—COMMON TYPE OF CLOSED FUEL NOZZLE. FIG. 8.—HESSELMANN FUEL NOZZLE.

piston, as are also the scavenging ports in the lower tier; while the scavenging ports of the upper tier are under the supplemental control of a timed, mechanically-operated valve. The lower tier and the mechanical valve open only after the exhaust ports have been uncovered long enough for the pressure in the cylinder to have fallen below the scavenging air pressure; the mechanical valve remains open after the exhaust ports have been closed. This arrangement not only prevents the dangerous blowing back of hot gases into the scavenging air passages, but also insures more thorough scavenging and the building up of a slight excess pressure in the pure air in the cylinder at the commencement of the compression stroke, thus enhancing its volumetric efficiency and power capacity.

Stationary Diesel engines may be roughly divided into slow-speed and medium-speed machines. High-speed engines of this type are confined to special purposes, such as the propulsion of submarine vessels. Without material variation in piston velocities, the speeds of both classes of stationary engines have been, more or less, selected to conform to the speeds of standard engine-type electric generators. The higher rotative speeds that are becoming general in Europe have, up to the present, not found favor with the American builder, despite the fact that they appear to be entirely successful. The following table compares the speeds of well-known European and American Diesel engines of about 500 H.P.:

	European.	American.
R.p.m.	240	150-200
Piston speed	4.15 m. per sec.	3.8-4.15 m. per sec.
Piston speed	815 ft. per min.	740-815 ft. per min.
Ratio, diam. to stroke	1 : 1.13	1 : 1.34-1 : 1.28

HORIZONTAL VS. VERTICAL ENGINES.

Within these limits there does not appear to be any appreciable difference in efficiency, reliability or wear in engines of equally good design and construction.

In Europe there has been a very general adherence to the vertical arrangement of Diesel engines, and of the few builders of horizontal machines the majority build vertical engines also. In America both styles have their champions, but at the present time the greater number appears to be following the lead of Europe.

There are comparatively few double-acting engines of the Diesel type in operation in Europe and considerably fewer in America. Practically all of these are horizontal machines. It may be reasonably asserted that there is not yet available a sufficiently extensive practical experience with this style to warrant the assurance of success in everyday service; and that, in spite of the many advantages which such an engine would possess, especially in large units, its development still lies well in the future.

Vertical Diesel engines are being built both with "A" frames and with inclosed crank cases. Coincidently with the tendency toward higher speeds, there has developed in Europe an increasing inclination toward the inclosed crank case; and Sulzer Bros., of Switzerland, recently exhibited in Berne a Diesel engine of this construction, of 1,000 B.H.P. rated capacity. The inclosed crank case for Diesel engines was adopted in America long before it received any consideration in Europe, and the original American builders still adhere to this style of frame for engines of small and medium capacities.

Practically all Diesel-engine cylinders are now built up of a liner, or inner barrel; a jacket, or outer cylinder, and a cylinder head. The space between the liner and the jacket forms the water jacket. The purpose of this construction is less that of facilitating the renewal of the inner barrel should it become worn and of permitting its free axial expansion,

than to enable the use of the most suitable iron to withstand the temperatures to which the cylinder is subjected, without the manufacturing risks involved in casting.

The admission and exhaust valves, which are exposed to the high combustion temperatures and to the corrosive action of the fuel, most of which contains sulphur, are now generally made up of a head of hard cast iron, with a stem of forged steel. In this country Diesel engines are not yet being built in sizes that would make the water cooling of the exhaust valves either necessary or convenient, but in the medium sizes the exhaust-valve stems generally work in water-cooled guides.

Few of the fuel oils in ordinary use are of such character that perfect combustion can always be attained under all conditions of load; it is therefore desirable that at least the exhaust valve, which is the most prone to fouling, may be dismantled with the greatest promptness and ease and with the least disturbance of adjustments.

More attention has been paid to the correct construction of the fuel atomizers than to any other single consideration, and with good reason. The experiments carried out on the first Diesel engine proved that the atomizing of the fuel and its introduction into the cylinder could best be performed by means of compressed air. The elements employed consist of an atomizer, an injection valve and a nozzle. These may be arranged to form the "closed" nozzle as originally developed by Diesel or the "open" nozzle as developed by Lietzenmayer. In the former the injection valve is placed between the air and fuel passages entering the atomizer and the nozzle inlet to the working cylinder, the valve thus separating both air and fuel from the cylinder until the time of injection. In the latter the injection valve is placed between the air passage entering the atomizer and the fuel inlet to the atomizer.

The "closed" nozzle construction shown in Fig. 7 is that which has attained the most widespread use. The atomizing elements consist of a series of finely perforated plates, the holes in the alternate plates being offset with reference to each other; followed by a slotted cone leading to the injection valve and to the nozzle plate at the entrance to the cylinder. Prior to the moment of injection the requisite quantity of fuel oil for the next working stroke is delivered above the perforated plates and distributes itself over these, the interior of the atomizer being in open communication with the compressed air. When the valve is lifted from its seat the air flows in a rapid stream over the plates and through the perforations, carrying the fuel with it and tearing it into fine globules. In its passage through the nozzle plate, after leaving the slotted cone, the mixture of air and oil atoms is deflected into a flat "umbrella," spreading over the surface of the piston.

The atomizer illustrated in Fig. 8 is also of the "closed" type and is the invention of Mr. Hesselmann, of the Swedish Diesel Engine Co. The typical feature of this is the siphonic arrangement of the passage in which the fuel lies between the time of its entrance into the atomizer and the opening of the injection valve, the atomizing taking place at the inner upper edge of the siphon passage.

The use of the "open" nozzle, as illustrated in Fig. 9, is confined, in America, to horizontal engines. In this the fuel is delivered by the fuel pump into a passage which, through a nozzle, is at all times in open communication with the cylinder. The compressed injection air is closed off from this passage by the injection valve. When this valve is lifted from its seat the stream of air scours over the surface of the accumulated fuel and atomizes it with an action similar to that of a file upon a metal surface. The final atomizing and spreading is performed in the passage through the nozzle.

As the load decreases and the quantity of fuel injected into the cylinder for each working stroke diminishes, the too rapid injection of the fuel

results in a more or less explosive ignition. To avoid this it becomes necessary to reduce the pressure of the injection air; that is, to slow down the injection velocity by reducing the pressure difference between the injection air and the air compressed in the working cylinder. In the cases of the "closed" nozzle this is accomplished automatically, to a limited degree; there being less fuel in the atomizer to retard the flow of air, more air is injected, and if the free-air capacity of the compressor remains constant, the injection pressure falls to correspond to this greater delivery volume. But this self-adjustment is insufficient to accomplish the purpose and has the additional disadvantage that, in spite of the more explosive ignition, the combustion is imperfect, because of the cooling effect of the excess air following the fuel into the cylinder, and it is necessary either to blow off a portion of the compressed air or to regulate the volume handled by the compressor. The latter is the general practice and is effected by throttling the suction of the compressor. This is usually done by hand, but some engines are provided with mechanical means operated by the governor.

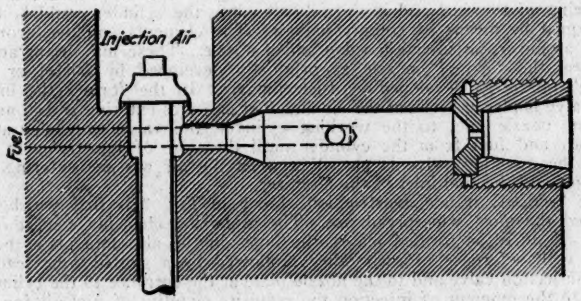


FIG. 9.—OPEN TYPE OF FUEL NOZZLE.

All engines of the Diesel type are started by compressed air. To avoid excessive use of air and to simplify the mechanism, it is usual to provide only one or two of the cylinders of a multicylinder engine with starting valves and to relieve the compression in the other cylinders until the engine has attained sufficient speed to effect this compression without risk of stopping. In small and medium-sized engines the relief of compression is generally accomplished by holding the exhaust valve partially open, which was formerly somewhat crudely done by slipping a yoke over the exhaust-valve lever; but now it is usual to mount the lever hubs on eccentric sleeves operated by hand levers, the arrangement being such that the rotation of the eccentric sleeve simultaneously places the fuel valve out of action and holds the exhaust valve open, and vice versa. This form of compression relief compels a separate handling of each cylinder and does not relieve the cylinders that are used for starting; it therefore becomes less convenient, less prompt and less effective as the size of the engine increases. For the larger sizes a system has been adopted in which auxiliary cams are used to operate the exhaust valves of all cylinders, opening these at the commencement of the compression stroke and closing them upon its completion; the fuel valves being meanwhile out of action until the engine has been accelerated sufficiently, when, by the operation of a single wheel or lever, the relief is cut off all the cylinders and the fuel is then admitted to them one at a time.

The construction of the pistons and their accessories has been reduced

to an almost uniform basis. In the larger sizes some builders provide the pistons with separate heads secured to trunk guides. Plain snap rings of cast iron have become universal, these being provided with ordinary lap joints. The piston rings are usually not doweled, to maintain fixed relative positions of the joints; except in the case of two-stroke cycle engines, in which the rings over-travel ports in the cylinder wall and it is necessary to prevent the joints from traveling across the openings. When the pistons attain diameters in which the area exposed to the heat becomes so great that radiation cannot be relied upon to prevent excessive temperatures, it is usual to positively cool the heads. This is particularly the case in two-stroke cycle engines, in which the pistons are exposed to combustion temperatures twice as often as in four-stroke cycle machines. As a cooling medium, either water, a mixture of water and air, or oil, is used.

REGULATING AMOUNT OF FUEL.

Of importance equal to that of the fuel-atomizing elements is the regulation of the amount of fuel delivered to the cylinder, to suit the existing load. The quantity, even at full load, is so minute that an almost infinitesimal variation will materially affect the speed of the engine. For example, the quantity of fuel oil delivered to a 100-H.P. cylinder, per working stroke, is only 0.0046 liter (0.28 cubic inch) at full load, and 0.0026 liter (0.16 cubic inch) at half load; or a difference of 0.002 liter (0.12 cubic inch) between full and half load. This fuel oil is delivered to the atomizers by means of a fuel pump having one plunger to serve each working cylinder; or one plunger to serve several cylinders, with a proportioning distributor to equalize the delivery to each to suit the average load conditions. Some of the methods used to regulate the fuel supply under the control of the governor are: Varying the stroke of the pump plunger; maintaining a constant stroke of the pump plunger, but bypassing back to the suction during the delivery stroke a variable quantity of the fuel; or maintaining a constant stroke of the pump plunger, with a constant pump-barrel volume during the suction stroke, and varying the pump-barrel volume during the delivery stroke.

The pressure of the air used for atomizing and injecting the fuel varies with the type of the engine and the proportionate loading. It ranges from 35 or 40 atmospheres at no load to 65 or 70 atmospheres at full load, a slight variation being desirable to suit the character of the fuel oil. This air is generally compressed by a three-stage air compressor integral with, or directly coupled to, the engine. To avoid lubrication difficulties and the danger of explosion of lubricating-oil gases, as well as to reduce the dimensions and power consumption of the compressors, they require thorough water cooling and the provision of ample inter- and after-coolers. Single-acting compressors with the trunk pistons arranged in echelon have become general, as also have automatic valves of very limited lift. As previously described, the capacity and delivery pressure of these compressors is adjusted by throttling the low-pressure suction.

LUBRICATION OF DIESEL ENGINES.

The main bearings of horizontal Diesel engines are generally constructed similarly to those of horizontal gas engines, with at least partial adjustability; whereas adjustable bearings have practically been abandoned in the modern makes of vertical engines, in favor of rigid bearings of such ample surface and provided with such efficient lubrication that their wear is negligible.

For cylinder lubrication it is desirable to use an uncompounded neutral mineral oil with a flash-point not lower than 180 degrees C. (350 degrees F.), a viscosity as high as the lubricating pump will handle reliably and

the characteristic of retaining a high degree of viscosity when hot. The function of this oil is not only to lubricate, but also to serve as a packing medium by forming a seal between the piston rings, against the leakage of air or gases. The compressor pistons are generally lubricated by a pump or by high-pressure sight-feed lubricators and use an oil similar to that for the working cylinders, except that a slightly lower viscosity and higher flash point are preferable.

In vertical engines the prevailing method of lubricating the bearings and pins is the forced-feed system, although some builders still adhere to ring oiling for the main bearings and centrifugal oiling for the pins.

To those accustomed to steam engines or, still more so, to steam turbines, the consumption of lubricating oil by a Diesel engine usually appears high, but this is the only operating expense on account of which the machine can possibly be accused of extravagance. This consumption averages 0.3 liter (0.07 gallon) per hour per 100 H.P. engine rating. Of this, about 20 to 25 per cent. is used in the cylinders.

The exhaust gases leave the cylinders at 670 to 700 degrees F. at full load, but the temperature rapidly falls with decreasing loads. To render the operation of the engine safe and comfortable, it is usual to water jacket such portions of the exhaust headers and pipes as the operator might come in contact with. Through these jackets is circulated a portion of the water that has already done duty in other parts of the engine where the rise in the temperature of the water must be maintained at a lower point. The exhaust gases from a properly designed, adjusted and operated Diesel engine are invisible and remarkably free from combustible constituents.

The majority of crude and fuel oils contain appreciable quantities of sulphur. This, if not over 1½ per cent., will not seriously affect any engine parts other than the exhaust passages and, to a less degree, the exhaust valves; but even in small amounts in combination with the hot vapors of the water of combustion, it will attack the exhaust pipes near the engine. For this reason, these pipes are made of cast iron and the use of steel for this purpose is avoided.

The noise of the exhaust is muffled by the same means as are in common use with gas engines; but concrete exhaust-muffling chambers have been found undesirable, on account of the disintegrating effect of the hot oil vapors, which may temporarily pass from the cylinders by reason of some maladjustment of the valve gear.

OILS SUITABLE FOR DIESEL ENGINES.

Fuel oils for use in Diesel engines should contain the lowest proportions of the following impurities, compatible with the prices demanded.

Water: More than ⅛ per cent. of water should be considered excessive, and if fuel containing more than this has been accepted, the water should be settled out by heating with a steam coil.

Sulphur: If in excess of 1½ per cent., the combination of the sulphurous fumes with the vapors of the water of combustion will corrode and pit the exhaust valves and seats and rapidly eat the exhaust piping.

Ash: A comparatively minute percentage of entirely non-combustible matter in the fuel causes, on the oily cylinder walls between these and the piston, an accumulation that will result in excessive wear.

Asphaltum: This much-abused term is susceptible to so many and various interpretations that it has no definite significance. Nor has the method for its determination been standardized. The various chemical and mechanical (penetration) determinations have little or no bearing upon the real issue under consideration, namely, the complete combustibility in a Diesel cylinder under the conditions existing in it and within the available time. A comparison of results obtained in actual use, with oils

containing a substance other than ash, which will not volatilize under certain definite conditions, appears to be the best guide as to the proportion of this substance that will render necessary an excessively frequent cleaning of the cylinder and its adjuncts. Several years of careful observations and records have induced the oldest Diesel-engine builders in this country to adopt, for such comparisons, the percentage of residue remaining after the sample of oil has been gradually brought to a temperature of 300 degrees C. and then subjected to this temperature for 120 hours in a closed furnace in which combustion does not take place. Under this treatment the sample is reduced to practically constant weight. It has been determined that, so long as this residue is less than 10 per cent. of the original weight of the sample, unreasonably frequent cleaning

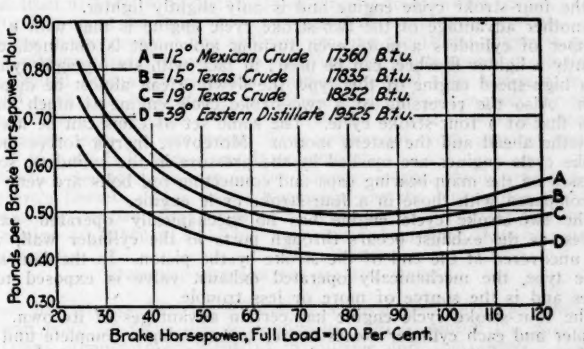


FIG. 10.—FUEL CONSUMPTION CURVES FROM ALLIS-CHALMERS ENGINE.

is not necessary. This percentage is equivalent to anywhere from 7 to 30 per cent. of "asphaltum," according to the various methods of determination in use. If the fuel oil contains more than 10 per cent. of residue, its use must be guided by the relative cost of the oil and the cost and inconvenience of labor and stoppages required for the more frequent cleaning. The form of the atomizer does not appear to have any bearing upon this question, as the substance does not become objectionable until after the fuel has entered the cylinder and its more volatile constituents have become gasified although it may render necessary warming of the fuel.

Fig. 10 illustrates the consumption of the fuel oils of widely varying gravity, in the same engine.—"Power."

TWO- VS. FOUR-STROKE CYCLE MARINE DIESEL ENGINE.

Some of the relative advantages of two- and four-stroke cycle Diesel engines for marine work and their present limitations in size and speed are contained in a paper by G. C. Davison, of the New London Ship and Engine Co., read at the recent International Engineering Congress in San Francisco. The following is in substance the author's remarks on this subject:

It is apparent that there exists today a certain difference of opinion as regards the merits of the two- and the four-stroke cycle Diesel engines.

While there are more firms building two-stroke cycle engines, at the same time those building four-stroke cycle engines seem to have met with more success. The two-stroke cycle engine superficially has the advantage of developing double the power for a given size of cylinder. It gives one impulse each revolution, while a four-stroke cycle engine requires two revolutions for one impulse. It may seem that the former should therefore furnish twice the power for an engine of given size. This, however, is far from realization. In order to operate on the two-stroke cycle principle a scavenging air system has to be provided. This requires considerable space and weight and a large number of additional working parts. The net result is that after adding the scavenging system the two-stroke cycle engine occupies nearly the same volumetric space as the four-stroke cycle engine and is only slightly lighter.

Another advantage of the two-stroke cycle engine is that with a given number of cylinders a more even turning movement is obtained, consequently a lighter flywheel can be used; in fact, with six or eight cylinders in a high-speed engine of this type, the flywheel can almost be dispensed with. Also the reversing of a two-stroke cycle engine is much simpler than that of a four-stroke cycle. The same set of cams can be used for both the ahead and the astern motion. Moreover, inertia forces in two-stroke cycle engines are masked by the pressure in the cylinders, so that stresses on the main-bearing caps and connecting-rod bolts are very small as compared with those in a four-stroke cycle engine.

The two-stroke cycle engine has no mechanically operated exhaust valves, as the exhaust occurs through ports in the cylinder wall, which are uncovered at the end of the stroke by the piston. In the four-stroke cycle type, the mechanically operated exhaust valve is exposed to hot gases and is the source of more or less trouble.

The four-stroke cycle engine has certain advantages of its own. It is simpler and each cylinder comes nearer to becoming a complete unit than in the case of the two-stroke cycle. As regards the number of parts, it has not only a less number of different parts, but also a less number of total parts, making it mechanically simpler. Also it is far superior in the heat conditions, as it handles only half the amount of heat developed in a two-stroke cycle engine of the same dimensions. Consequently, for cylinders below a certain size it does not require artificial cooling of the piston, as is necessary in the two-stroke cycle type. In the larger sizes piston cooling is necessary in both types, but the four-stroke cycle has the advantage of working with a much lower mean temperature, consequently there is less likelihood of cracks due to heat conditions. Moreover, it is more economical than the two-stroke cycle.

LIMITATIONS IN ENGINE SIZES.

There are certain limitations to the size of Diesel engines which may be advantageously applied to marine propulsion. The smallest than can well be used is in the neighborhood of 100 H.P. As compared with gasoline engines the first cost of a Diesel of small power is such that below this limit the saving in cost of fuel is sufficient only to offset the interest and depreciation. As the size increases, the economy in fuel becomes of greater importance. The upper limit of size is at present fixed by the designs and materials now in use and is approximately 400 H.P. per cylinder in single-acting units. That is, this represents about the upper limit of engines that have been built and are in successful operation. Experiments have been made, however, on double-acting two-stroke cycle engines developing 2,000 H.P. per cylinder, which represents about the theoretical limit unless some radical departure is made.

Marine Diesel engines may be roughly divided into two general classes, according to weight per horsepower. The light-weight class is repre-

sented by the engines used in submarines, which run at piston speeds from 1,000 to 1,200 feet per minute. A 1,000 H.P. 6-cylinder engine will in some cases run as high as 450 r.p.m. and will weigh approximately 40 pounds per H.P.

A heavy-duty slow-speed engine developing 1,000 H.P. at 100 r.p.m. will weigh as much as 300 pounds per H.P. Intermediate types are also built; for instance, the 1,000-H.P. Nuremberg engine of the U. S. S. *Fulton*, running at 260 r.p.m., weighs 100 pounds per H.P.

About 300 Diesel engines were built for submarines previous to the present war; since the war began a large number have been built or are building, the sizes ranging from 300 to 1,000 H.P. Destroyers require such enormous power (from 10,000 to 15,000 H.P.) that it is not practical, at least at present, to install Diesel engines to give this output. Certain foreign countries have, however, installed both steam turbines and oil engines in destroyers. For ordinary cruising the Diesel engine is used, thereby effecting great economy in fuel, and for high speed, the turbine.

The United States Navy is now building the fuel ship *Maumee*, to be equipped with two Nuremberg two-stroke cycle Diesel engines, developing about 2,500 H.P. each. These will be the largest slow-speed heavy-duty engines thus far attempted. For commercial work marine engines from 50 to 2,400 H.P. are in regular operation.

A further development, yet in its infancy, is the double-acting two-stroke cycle Diesel. Prior to the European war the Nuremberg Co. had been working for over four years on a 6-cylinder double-acting two-stroke cycle engine of 12,000 H.P. This was reported to have been completed last summer.

ECONOMIC OUTLOOK FOR MARINE DIESELS.

There has always been a difference between the European and the American point of view, due to conditions. Up to within the past year capital was comparatively plentiful in Europe and fuel scarce. Consequently the European ship-owner considered ultimate saving and was willing to pay a greater first cost for his propelling plant if the operating economy would show an ultimate gain. In the United States the shipping business has never been given much encouragement and those who have gone into the business have had to consider first costs very seriously. Furthermore, both coal and oil are comparatively cheap in this country. Finally, a further drawback to American development has been the lack of trained operators. In the course of time the basic advantages will be realized in this country and the necessary trained operators will be developed. Under present circumstances it is impossible to install a Diesel-engine plant at the same cost as a steam plant. There is a possibility that there may eventually be developed a Diesel engine which will cost so little more than a steam plant that the difference will not be worth serious consideration. When that time comes we shall probably see a boom in the marine Diesel-engine business.

Probably the worst enemies of the marine Diesel engines during the past few years have been its over-enthusiastic advocates, many of whom have made promises that they could not fulfill. Others have built and installed engines that were nothing but experiments. New firms are continually entering the field, little realizing that the design and construction of these engines are highly developed specialties. The first ones produced in this way are generally failures and unfortunately the good and the bad suffer as a result.—"Power."

HIGH-SPEED, SUPERHEATED STEAM TURBINE BLADING.

An interesting contribution to the subject of so-called non-ferrous metals and alloys used for turbine blading was given in a paper presented to the Institute of Metals on the 17th September by Mr. W. B. Parker, F. I. C., in which an exposition of the present position and of the trend of future advancement was summarized. The author also demonstrated the importance of the subject, both industrially and nationally, and advocated the scientific control of manufacture and original research.

Mr. Parker suggested that particulars of turbine designs should be interpreted into metallurgical equivalents, such as working conditions and physical and chemical properties, and showed how these are best defined in material specifications. Terms commonly used in purchasing specifications covering the physical properties were reviewed and criticised, and practical suggestions were given respecting definitions for these terms, with suitable methods for determining the properties they cover. In reviewing the types of non-ferrous materials already tried, the author stated that pure brasses of the α, $\alpha + \beta$ and β types, as well as numerous so-called manganese brasses, manganese bronzes, and other complex copper-zinc alloys, have proved unfit for modern conditions of high-speed turbines driven with superheated steam.

Copper, nickel, cobalt and the alloys of copper-nickel, copper-manganese, copper-aluminum, copper-aluminum-manganese, copper-aluminum-nickel and phosphor-bronze were critically considered by Mr. Parker, and the results of original tests made by him were given. He pointed out that, although non-ferrous materials have the advantage of non-rusting properties, at present alloy steels were utilized for the highly-heated and stressed-rotating blades because of the necessity of having a material which possesses *naturally* a good "proportional limit" which is stable, *i. e.*, not seriously lowered by prolonged exposure to highly superheated steam. None of the non-ferrous materials at present in use fulfil the latter condition, and the aim of the non-ferrous manufacturers must be, therefore, to produce a material in a physically stable condition, that is, in the naturally soft or annealed condition. This material should possess a "proportional limit" of over 16 tons per square inch, a tensile strength exceeding the proportional limit by not less than 100 per cent., and an elongation not less than 10 per cent. The proportional limit should remain constant within 10 per cent. of its "cold" value over the range of temperature from 212 degrees to 844 degrees F.

In conclusion, the author gave further particulars of a tentative specification, and schemes for researches for materials to comply with the same."—"The Shipbuilder."

HARDENING AND TEMPERING HIGH-SPEED TOOL STEEL.

THE EFFECT OF CHROMIUM AND TUNGSTEN UPON THE HARDENING AND TEMPERING OF HIGH-SPEED TOOL STEEL.*

By Professor C. A. Edwards, D.Sc., and H. Kikkawa, Manchester University.

The only systematic British investigation which has been published regarding the tempering of high-speed cutting steels is that by Professor Carpenter (1, see references). The experiments which he describes were made with two series of steels—namely, chromium-molybdenum and chromium-tungsten. The latter series, consisting of five speci-

*Paper read before the Iron and Steel Institute, September 23, 1915.

mens, are all that need be considered in connection with the present work. Their compositions are given in Table I.

TABLE I.

No.	Carbon.	Silicon.	Chromium.	Tungsten.
	per cent.	per cent.	per cent.	per cent.
12	0.98	0.24	3.1	7.96
13	0.77	0.29	3.70	10.83
14	0.85	0.15	3.0	12.5
15	0.63	0.13	2.2	12.8
16	0.55	0.15	2.5	13.5

In determining the temperatures at which tempering took place, the method adopted by Professor Carpenter consisted in heating the hardened specimens for one hour at the desired temperature and then allowing them to cool in the furnace. After this treatment the sections were polished and examined under the microscope, and the tempering judged by noting the changes of structure from the polygonal austenitic crystals through the various stages towards troostite, etc., but no hardness determinations were made. Briefly, the conclusions he arrived at were as follows: "No tempering occurs below 550 degrees C.; all undergo an appreciable tempering at 550 degrees C. No. 12 is fully tempered at 600 degrees C. No. 15 at 650 degrees C., and Nos. 13, 14 and 16 at 700 degrees C." Experiments were also made to ascertain the effect of the initial hardening temperature upon the tempering of these steels, from which it was concluded that "if the alloy No. 14 be quenched below 1,100 degrees C., tempering is already well advanced at 550 degrees, and is complete at 600 degrees C.; if it be quenched at 1,150 degrees or above, tempering is less advanced at 500 degrees C., and is not complete till 650 degrees C.; but the results at 1,100 degrees, 1,150 degrees and 1,200 degress are all about the same."

Valuable as these results undoubtedly were, it soon became evident that they were inconclusive, for in discussing them along with the cooling and heating curve data which were also obtained by Professor Carpenter (2), Mr. Taylor (3) concluded that they did not provide a substantial indication of the relative cutting properties of high-speed steels. His reason for this view was that he had found, as a result of many carefully conducted experiments, that the best cutting properties were obtained after subjecting tools to a secondary heat treatment at a temperature above that for which the heating curves and microstructures seemed to show that complete tempering would occur. In consequence of this Mr. Taylor said: "These facts indicate clearly that the two methods as yet devised by scientists for determining the most important quality of the new high-speed steels are ineffective," and he was of the opinion that the only satisfactory method of deciding upon the relative merits as regards cutting was to make elaborate standard cutting-speed tests with carefully standardized tools upon standardized forgings. But he says: "This test requires so much expensive apparatus, consumes so much time, and is so slow, that a simpler index or guide which will indicate correctly the quality of high-speed tools is much needed. Moreover, we firmly believe that in time some simpler index to the property of 'red-hardness' in tools will be found."

The second heat treatment which Mr. Taylor found so beneficial consists in heating the tools, which have already been hardened, to a temperature somewhere between 370 degrees and 671 degrees C. The maxi-

mum permissible temperature for this secondary heating was not very rigidly fixed, but the following statement seems to indicate that it is somewhat lower than 671 degrees C. The second or low heat treatment consists of heating the tools—

"(*a*) To a temperature below 671 degrees, preferably to 621 degrees C., for a period of about 5 minutes; and

"(*b*) Cooling to the temperature of the air, either rapidly or slowly."

It was also pointed out that heating a little above this higher limit, even for a very short time, almost ruins the cutting properties of a tool.

From a consideration of the above facts, it is quite evident that further research upon the effect of chromium and tungsten on the tempering of modern cutting tools should be of considerable interest, and the results obtained of some practical value. Further, it would be useful to know the cause of the improved cutting properties which are brought about by the second heat treatment.

In a paper dealing with "The Function of Chromium and Tungsten in High-Speed Steels," one of the present authors (4) made a few preliminary experiments in this direction, and, judging from the evidence then obtained, it was thought that the failure of high-speed cutting tools was not due to the loss of hardness, but to the formation of a brittle constituent, which destroyed their cutting edge. It is, however, necessary to say that this conclusion is incorrect, for during the present investigation it has been found that the brittle constituent does not form on tempering steels which have been properly hardened. It can only be detected in material which has been overheated during the high heat treatment. The analysis of the steels which have been used are shown in Table II. It will be seen that these may be divided into two series: (*a*) Those with an approximately constant percentage of carbon and chromium but with varying tungsten, and (*b*) those with the same carbon and tungsten content and varying chromium. These two series are all that could be desired for determining the influence of chromium and tungsten upon the tempering properties of the steels.

TABLE II.

No.	Carbon.	Chromium.	Tungsten.	Silicon.
	per cent.	per cent.	per cent.	per cent.
1	0.63	6.15	...	0.07
2	0.65	6.13	3.08	0.05
3	0.67	6.04	5.92	0.08
4	0.65	6.19	9.12	0.04
5	0.67	6.18	12.50	0.06
11	0.63	...	19.28	0.07
12	0.63	1.12	19.40	0.06
13	0.68	3.01	19.37	0.04
14	0.64	4.91	19.33	0.09
10	0.67	5.99	18.86	0.10
9	0.64	6.24	17.69	0.07

In all cases the specimens which were used were 1 inch square and ¾ inch thick. They were hardened by heating in a crucible contained in an injector furnace to about 1,350 degrees C. for 10 minutes, followed by quenching in an air blast. In this way they were cooled to below a red heat in about 1¼ minutes. The hardened specimens were then carefully ground on a whetstone to obviate any tempering or change of structure, and finally polished for microscopic examination. If this examination re-

vealed a structure which indicated that the specimen had begun to melt, another sample was taken. In the hardened pieces the so-called austenitic or polygonal crystal structure was aimed at, but it was not possible to get this in steels with less than 3 per cent. of chromium or 9 per cent. of tungsten. The former contained numerous specks of undissolved carbide or tungstide, and the latter were more or less martensitic; in fact, No. 1, with no tungsten, was entirely martensitic.

All tempering experiments up to 800 degrees C. were made in an electric resistance furnace, the temperatures being carefully registered by means of a thermo-couple which was placed in contact with the specimens and connected with a portable pyrometer. For heating to temperatures above 800 degrees C. a gas-fired muffle furnace was used. In most cases the tempering consisted in heating for one hour after reaching the desired temperature and then allowing the pieces to cool on an asbestos pad in the ordinary atmosphere. After each treatment hardness determinations and microscopic examinations were made, and then the same specimens were heated at the next higher temperature. Preliminary experiments were first made with steel No. 9, because it contains approximately the same carbon, chromium and tungsten, as recommended by Mr. Taylor. The hardness data are given in Table III, and plotted in Fig. 1, page 1026. In this instance the hardness of the hardened material was 627, which fell to 596 after heating for two hours at 494 degrees C.; heating for 24 hours at the same temperature had practically no further effect. By heating to 542 degrees a substantial increase in the hardness is brought about, and a still further increase occurs at 589 degrees; in fact, after heating at the latter temperature the material is much harder than it was

TABLE III.—PRELIMINARY EXPERIMENTS. STEEL NO. 9. HARDENED IN AIR BLAST.

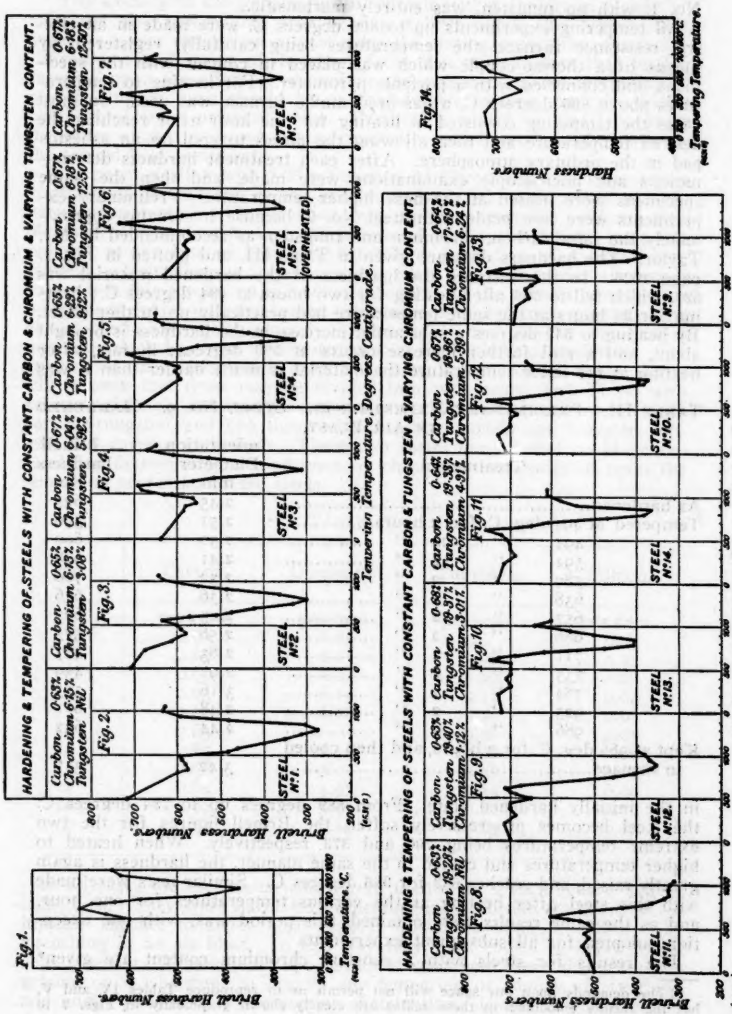
Treatment.	Indentation. Diameter. mm.	Brinell hardness No.
As hardened	2.45	627
Tempered at 494 deg. C. for 2 hours	2.51	596
494 " 24 "	2.50	600
592 " 2 "	2.41	648
589 " 2 "	2.32	700
638 " 2 "	2.36	676
657 " 2 "	2.45	627
686 " 2 "	2.56	573
711 " 2 "	2.83	467
735 " 2 "	2.92	439
784 " 2 "	3.16	373
925 " 2 "	2.48	611
986 " 2 "	2.44	632
Kept at 986 deg. C. for 2 hours, and then cooled in furnace	3.42	317

in the initially hardened state. From 589 degrees up to 784 degrees C. the steel becomes progressively softer, the Brinell figures for the two extreme temperatures being 700 and 373 respectively. When heated to higher temperatures and cooled in the same manner, the hardness is again greatly raised, and reaches 632 for 986 degrees C. Similar tests were made with this steel after heating at the various temperatures for one hour, and as the same results were obtained, this period was, with one exception, adopted for all subsequent experiments.

The results for steels with a constant chromium content are given*

*[The demands upon our space will not permit us to reproduce Tables IV and V, but the results embodied in these tables are clearly shown graphically in Figs. 2 to 13.—ED. E.]

in Table IV, and graphically illustrated in Figs. 2, 3, 4, 5, 6 and 7; and those for steels with a constant percentage of tungsten are shown in Table V, and plotted in Figs. 8 to 13. Broadly speaking, it will be seen that the effect of tempering is very similar in all cases for the series containing the same percentage of chromium. The differences, which must be attributed to the influence of tungsten, and which are certainly



important, are clearly variations in the degree of certain features that are characteristic of all the steels belonging to this group. In all cases the first effect of tempering is to bring about a marked softening; but after heating above certain temperatures which lie between 397 degrees and 589 degrees C., they again become harder. On examining the diagrams in Figs. 2 to 7, and allowing for some slight irregularities, it will be observed that the temperature at which this secondary hardening begins is progressively raised with increasing percentages of tungsten. This rise is almost exactly 50 degrees C. for each increment of 3 per cent. of tungsten up to 12.5 per cent., and then a further rise of 50 degrees C. for the next 6 per cent. of tungsten. Perhaps the most important point to note in this connection is the total increase of the secondary hardening. With the pure chromium steel this is comparatively small; it is only from 580 to 596 in Brinell figures. It is more pronounced (580 to 636) in the presence of 3 per cent. of tungsten, and still more so with 5.92 per cent. of tungsten. In fact, the net effect in the latter case is an increase of 130 in the Brinell figure, which makes the steel much harder after being heated to 589 degrees C. than it was in the initially hardened condition. Similar results were obtained with steels containing higher percentages of tungsten, and it is significant to note that the degree of secondary hardening becomes higher as the amount of tungsten contained in the steel is raised. As regards the temperature at which the maximum secondary hardness is obtained, which also corresponds to the temperature above which real softening or annealing begins, it will be noticed that for the steel with no tungsten this is 494 degrees; for the one with 3 per cent. of tungsten, 542 degrees C.; and 589 degrees for 5.92 per cent. of tungsten; but higher percentages do not appear to have any further influence in this respect. By heating above these critical temperatures softening occurs in all cases, and the completely annealed state is reached at 784 degrees C. After heating from 784 degrees to 1,000 degrees the steels again become harder, and this is an indication of what is generally regarded as the property of self-hardening, which has usually been attributed to the action of tungsten; but the data contained in Fig. 2 very clearly show that a pure chromium steel also possesses this property to a very remarkable extent.

Turning now to the results obtained from the series with a constant percentage of tungsten, which are illustrated in Figs. 8 to 13. The main features for those steels containing more than 3 per cent. of chromium are very much the same as those just described. Marked differences are, however, evident for the pure tungsten steel and the one containing 1.2 per cent. of chromium. In the former case the hardness after the high heat treatment and air-quenching is only 500, which is much lower than for the pure chromium steel. Tempering up to 494 degrees C. has very little influence, but from 494 degrees to 638 degrees C. the Brinell figure is raised to 610. With 1.2 per cent. of chromium the hardness falls only very little after heating up to 638 degrees C. It is a most curious fact that in both these cases the temperature at which real softening begins is 50 degrees higher than in any of the other steels, but it is necessary to remember that in the one without chromium the hardness never reaches such a high value as with those containing chromium. From 638 degrees to 884 degrees C. there is a rapid decrease in the hardness; but whereas in all other cases heating at from 884 degrees to 1,048 degrees C. is followed by a marked degree of self-hardening, in the pure tungsten steel this property is entirely absent under these conditions of cooling, and it is not very pronounced in steel No. 12.

In view of the fact that Mr. Taylor recommends a five minutes' heating for the secondary treatment, it was thought advisable to determine the effect of that period as compared with the results obtained after an hour's heating. This was done in the case of steel No. 9, and the results are

shown in Table VI and Fig. 14, from which it will be seen that the only difference is in the temperature at which the maximum degree of secondary hardening is obtained. After heating for five minutes, this temperature is 25 degrees C. higher than on tempering for one hour.

TABLE VI.—TEMPERING FOR FIVE MINUTES.

Tempering temperature. Deg. C.	Brinell hardness No.
As hardened.	708
207	719
302	700
397	659
445	676
494	665
542	708
589	719
614	732
638	708
662	648
686	627

As a result of the foregoing experiments it is possible to offer a more rational explanation of the function of chromium and tungsten in modern high-speed tool steels. Contrary to the opinion which has been held by some workers, it is now unquestionable that the element chromium imparts to steel the property of self-hardening by cooling in air from high temperatures. Indeed, a steel containing 6 per cent. of chromium, along with 0.63 per cent. of carbon, possesses this property in a much greater degree than one containing 18 per cent. of tungsten; it is not only much harder after the usual air-quenching from about 1,300 degrees C., but its hardness is progressively raised, even with comparatively slow rates of cooling, as the initial temperature is increased from 800 degrees to 1,000 degrees C. On the other hand, no hardening occurs when the tungsten steel is cooled under exactly the same conditions from those temperatures.

Briefly, then, the action of chromium in these steels is as follows:

(*a*) In conjunction with carbon, it is the cause of the great hardness of high-speed steels; and

(*b*) It produces a marked lowering in the temperature at which hardening can be effected.

As regards tungsten, its action in the absence of chromium is to raise the temperature at which tempering or annealing begins, and in the presence of chromium it increases the intensity of the secondary hardness which is brought about by the low heat treatment, and raises the tempering temperature.

There now appears to be very little difficulty in giving a trustworthy explanation of Messrs. Taylor and White's important discovery of the fact that a secondary low heat treatment improves the cutting efficiency of modern high-speed steel. When this treatment is conducted at the temperature they advise—namely, 620 degrees C.—it gives the maximum degree of hardness. Although in the authors' experiments the hardness was determined at the ordinary temperature, after the heating, there seems to be no doubt that the figures obtained in this way are a measure of what Mr. Taylor describes as "red-hardness." For the determinations which were made after heating for periods of five minutes, the maximum hardness obtained corresponds to 614 degrees C., which is practically the same temperature as Mr. Taylor suggests. These facts lead the authors to believe that valuable information as regards the relative merits of high-speed cutting-steel can be obtained from careful tempering experiments when made in conjunction with hardness determinations. Fur-

ther, when more data are accumulated this simple method may be found sufficient to replace almost completely the very elaborate and costly standard cutting tests which are now necessary. In any case it should be extremely useful for ascertaining the best temperature for the second or low heat treatment.

Effect of Initial Hardening Temperature.—Mr. Taylor has made it abundantly clear that the hardening temperature is a factor which has a most decisive influence upon the cutting properties of high-speed tools. He found that the higher the temperature, short of actual melting, the greater the cutting efficiency. This improvement is certainly not due to an increase in the hardness of the steel, because whenever differences are observed in this connection it is invariably found that a greater hardness is produced by air-quenching from the lower temperatures down to about 1,000 degrees C. Cooling curves which have been taken from temperatures between 1,000 degrees and 1,320 degrees C., as shown by Professor Carpenter (1) and one of the present authors (4), indicate that the initial temperature has a pronounced effect upon the character of the carbide change-point, but it would really be going too far to say they are sufficient to show that better cutting powers should be obtained by treating the tools from the higher temperatures. Hence with a view of getting more direct and decisive evidence respecting this aspect of the question, tempering experiments were made with steels which had been hardened by air-quenching from about 1,050 degrees C. Several steels were used for this purpose, but as the results were practically the same in each case, it is only necessary to refer to a typical example. The data given in Table

TABLE VII.

Tempering temperature. deg. C.	Indentation. Diameter. mm.	Brinell hardness No.
As hardened	2.25	744
302	2.35	683
397	2.38	665
445	2.39	659
494	2.39	659
542	2.37	670
589	2.51	596
614	2.64	535
638	2.73	503
686	3.01	415
735	3.22	359
784	3.29	343

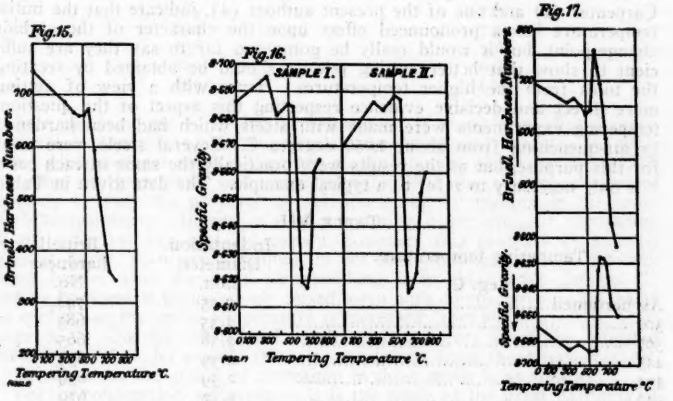
VII and plotted in Fig. 15 relate to steel No. 9. It will be noted that the hardness was in the first place 744, which is considerably greater than for the same steel when quenched from just below its melting-point, 1,350 degrees C. After tempering for an hour at 300 degrees C., the hardness fell to 683, and at 445 degrees C. to 659; it then remained the same for 494 degrees C., and increased slightly at 589 degrees C., but rapidly decreased at all higher temperatures. When compared with the tempering curve (Fig. 13) it will be readily seen that these results are in entire agreement with Mr. Taylor's discovery. They show that, although the steel is harder when quenched from 1,054 degrees C. than from 1,350 degrees C., it loses its hardness much more readily, and only gives a faint indication of the secondary hardening. This difference is very marked, for with the lower hardening temperature annealing begins at 542 degrees C., whereas in the other case the maximum hardness is obtained at 589 degrees C., and at those temperatures the respective Brinell figures are 670 and 756.

It is now possible to give a clear idea of why the tempering properties of these steels are so largely dependent upon the hardening temperature, but for this purpose it is necessary to remember the following facts:

1. The whole of the carbon and chromium of a steel containing carbon 0.63 and chromium 6.15 per cent. is in complete solution at all temperatures above the Ac critical points; and

1*a*. These elements can be retained in solution by air-quenching from any temperature above the carbide change, and when this is done the steel is as hard as water-quenched carbon steel.

1*b*. A hardened chromium steel of the kind under discussion only develops a comparatively slight degree of secondary hardening (see Fig. 2, page 1026), and the Brinell hardness drops from 720 to below 600 after the steel has been tempered at all temperatures above 300 degrees C.

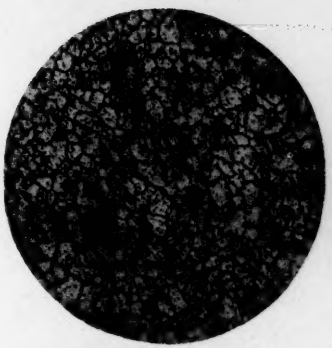


2. With a steel containing about 19 per cent. of tungsten and 0.63 per cent. of carbon the latter element goes into solution immediately the Ac points have been passed, but a very large proportion of the tungsten remains undissolved even when the temperature is raised to the melting point. This is shown in Photomicrograph No. I. It is not known in what form this undissolved tungsten exists, but it is most probably present as the compound Fe_2W, which has been isolated by Professors Arnold and Read (5).

2*a*. Very rapid air-quenching from temperatures up to 1,050 degrees C. has practically no hardening effect upon the above tungsten steel, and even when quenched from about 1,350 degrees C. it is much softer than a similar specimen of chromium steel which had been slowly cooled in air. (See Table VIII.)

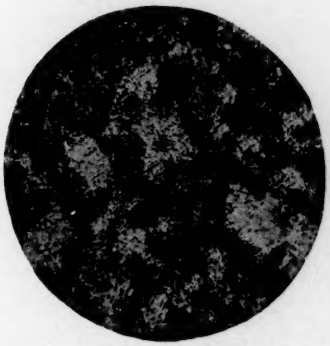
2*b*. But when tungsten steel is tempered it undergoes a very pronounced secondary hardening, and, in fact, after being heated to 638 degrees C., it is harder than the chromium steel which has been tempered at any temperature above 300 degrees C.

3. The presence of chromium in high-speed steel increases the solubility of the tungsten to such an extent that with 6 per cent. of the former element, 19 per cent. of the latter can be held in solution at about 1,350 degrees C.



No. I.—Chromium, 1.12 per cent.; Tungsten, 19.14 per cent. Air-quenched from 1350 deg. Cent. Magnified 150 diameters and slightly reduced.

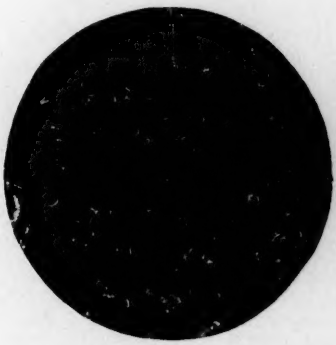
No. II.—Steel No. 9. As hardened. Magnified 150 diameters and slightly reduced.



No. III.—Steel No. 9. Tempered at 494 deg. Cent. Magnified 150 diameters and slightly reduced.

No. IV.—Steel No. 9. Tempered at 589 deg. Cent. Magnified 150 diameters and slightly reduced.

Nos. I to IV.—Tempering of High-Speed Steels.



No. V.—Steel No. 9. Tempered at 638 deg. Cent. Magnified 150 diameters and slightly reduced.

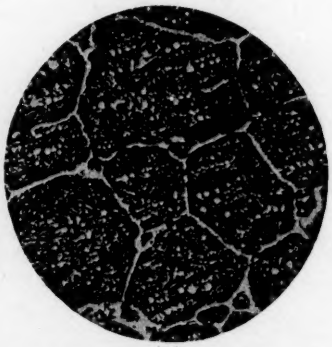
No. VI.—Steel No. 9. Tempered at 784 deg. Cent. Magnified 150 diameters and slightly reduced.

No. VII.—Steel No. 9. Tempered at 884 deg. Cent. Magnified 150 diameters and slightly reduced.

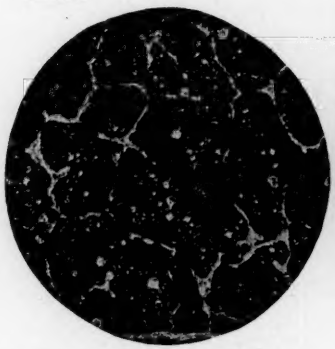
No. VIII.—Steel No. 9. Tempered at 986 deg. Cent. Magnified 150 diameters and slightly reduced.

Nos. V to VIII.—Tempering of High-Speed Steels.

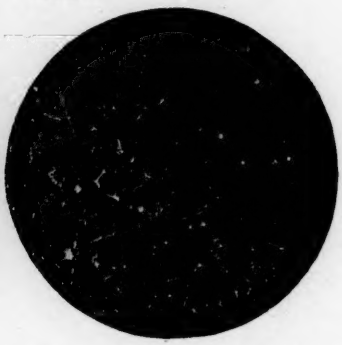
No. IX.—Overheated during hardening. Magnified 150 diameters and slightly reduced.



No. X.—Tempered at 302 deg. Cent. Magnified 150 diameters and slightly reduced.



No. XI.—Tempered at 445 deg. Cent. Magnified 150 diameters and slightly reduced.



No. XII.—Tempered at 784 deg. Cent. Magnified 150 diameters and slightly reduced.

Nos. IX to XII.—Over-Heated and Tempered High-Speed Steel.

TABLE VIII.—HARDNESS OF STEELS NOS. II AND 9 AFTER COOLING IN AIR FROM DIFFERENT TEMPERATURES.

Size of specimens, 1 inch by 1 inch by $\frac{1}{8}$ inch.

...	Temperature.			
	764 deg. C.	834 deg. C.	914 deg. C.	986 deg. C.
	Brinell No.	Brinell No.	Brinell No.	Brinell No.
No. II	268	282	286	313
No. 9	377	464	622	652

...	Temperature. 1058 deg. C.	Air-quenched from 1048 deg. C.
	Brinell No.	Brinell No.
No. II	313	354
No. 9	655	676

Bearing the above facts in mind, the authors believe that the influence of initial temperature may be correctly summarized as follows:

1. When the hardening temperature is low the tempering properties approximate to those of a pure chromium steel, softening takes place at lower temperatures, and little or no secondary hardening occurs. Under these conditions the full influence of the tungsten is not obtained, because a large proportion remains inert in consequence of it being out of solution.

2. The maximum resistance to tempering, and the greatest degree of secondary hardening, can only be obtained by getting the tungsten in complete solution, and for modern high-speed steels this can only be effected by heating the steel to about 1,350 degrees C.

Microstructures.—Since the microstructures of these steels are in many ways so remarkably similar, and as their chief features have been previously described (2 and 4), it is only necessary to refer to the structural changes that occur after they have been tempered at the various temperatures. For this purpose all that is needed is to describe the structures of one steel, which may be taken as typical of the series. No. II (facing 1030) represents the structure of steel No. 9 in the hardened state, and shows the well-known polygonal austenitic crystals. The structure of the same material after being tempered at 494 degrees and 589 degrees C. is quite martensitic (Nos. III and IV); this corresponds to the appearance of the secondary hardening. After being heated to 638 degrees C. (No. V), specimens are more readily etched, and show signs of the decomposition of the martensitic structure, but the martensitic markings are still evident, and are not completely removed until a temperature of between 700 degrees to 750 degrees C. is reached. After being heated to 784 degrees C. the material becomes fully annealed, but the structure No. VI shows irregular-shaped patches of carbide. When heated to 884 degrees C. and cooled, hardening again sets in, but the structure No. VII is so remarkably granular and fine as to make it impossible accurately to describe it. There can, however, be very little doubt that it is partly martensitic, because at

the next higher temperature it becomes almost purely martensitic. (See No. VIII.)

Steels which have been overheated or partly melted show unmistakable signs of this treatment. The structure of a specimen of this kind is shown in No. IX, where the broad crystal boundaries and globular patches in the interior of the crystals represent portions of the steel which were liquid at the temperature from which they were air-quenched. From Nos. X, XI and XII, it will be seen that the white background passes through the same structural changes on being tempered as a properly hardened specimen, but the partly melted portions are very persistent, and not entirely removed after heating for an hour at 784 degrees C. Steels which have been overheated are invariably brittle, and are liable to crack under the Brinell test, in which case the fracture follows the crystal boundaries.

Hardness and Specific Gravity.—When metals are hardened, either by mechanical work or by heat treatment, changes of volume usually occur, and considerable importance is now attached to these changes, because they are supposed to indicate the cause of the hardening. The present paper would therefore be incomplete without some reference to this question.

For this purpose it was thought that the best results would be obtained by first hardening the steel, then, after carefully grinding and polishing each surface of the specimen, determine its specific gravity, and repeat these determinations on the same samples after it had been tempered at the various temperatures for an hour. The object of using the same sample was to eliminate all possible errors that might otherwise be introduced by slight variations in the composition, or other defects, in different samples. Two independent series of experiments were made with steel No. 9. The results that were obtained are given in Table IX, and plotted

TABLE IX.

Tempering temperature.	Specific gravity.	
	Sample No. 1.	Sample No. 2.
Deg. C.		
0	8.686	8.670
302	8.695	8.686
397	8.681	8.683
445	.....	8.685
494	8.685	8.688
542	8.684	8.687
589	8.624	8.627
614	8.619	8.616
638	8.617	8.617
686	8.624	8.622
735	8.660	8.652
784	8.664	8.660

in Fig. 16. The chief characteristics of the two curves are fundamentally the same; indeed, with the exception of the values in the hardened state and those after tempering at 300 degrees C., the results are virtually identical. In considering the details of these results, it will be seen that the effect of tempering is first to increase the density or specific gravity of the mass; at 397 degrees C. there is a slight decrease, which, in turn, is

followed by another small increase at 445 degrees, 494 degrees, and 542 degrees C.; but the most striking discontinuity occurs between 542 degrees and 614 degrees C. This change corresponds with a very remarkable decrease in the density which takes place at precisely the same temperature range as that of the secondary hardening. Above 614 degrees C. —that is, when the steel begins to be annealed—the density is again raised. When these results are compared with the tempering hardness curve for the same steel, Fig. 13, the authors venture to say that the most casual observer cannot help being impressed by the resemblance which undoubtedly exists between the graphical illustrations of the two sets of experiments. Indeed, it is almost impossible to resist the conclusion that there is a direct connection between hardness and volume. This will perhaps be more evident on examining Fig. 17, where the density and hardness data are reproduced in such a way as to give diagrams of approximately the same size. From these curves it will be seen that each increase of hardness is accompanied by an increase in volume, and vice versa. The authors do not propose at this stage to utilize these extremely interesting facts to support any particular theory of hardening steel. One of the reasons for doing this is that they are quite consistent with at least two of those theories. The essential points to remember are:

1. The greatest hardness is associated with the highest volume.

2. When metals are hardened by mechanical deformation, similar volume changes are obtained.

Conclusions.—The conclusions which have been drawn from this investigation are:

1. The first effect of tempering hardened high-speed steels is to make them softer, but when they are tempered at higher temperatures they again become harder, and after heating at or about 614 degrees C., they are much harder than in the initial air-quenched state. There can be no doubt that this secondary hardening is the cause of the improved cutting powers of a tool, which Mr. Taylor found was brought about by a secondary low heating to about 620 degrees C.

2. Chromium in conjunction with carbon is the cause of the great hardness of hardened high-speed steels, and further, it materially lowers the temperature at which these steels can be air-hardened.

3. In the absence of chromium, tungsten raises the temperature at which tempering or annealing begins, and in the presence of chromium it increases the intensity of the secondary hardening, and raises the tempering temperature.

4. Tungsten steel containing 18 per cent. of tungsten and 0.63 per cent. of carbon can be air-hardened only by rapid air-quenching from temperatures above 1,050 degrees C.

5. When high-speed steels are hardened at low temperatures, say 1,050 degrees C., the tempering properties approximate to those of a pure chromium steel, by softening at a low temperature, and developing little or no secondary hardening. This is due to the tungsten being undissolved and remaining inactive.

6. The maximum resistance to tempering and the greatest degree of secondary hardening can only be obtained by getting the tungsten into solution, and with modern high-speed steels this is not complete until a temperature of about 1,350 degrees C. is reached.

7. Specific gravity determinations seem to indicate that there is a direct connection between the hardness and volume of these steels. On tempering, an increase of hardness is accompanied by an increase in volume.

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HIGH-SPEED REDUCTION GEARS.

During the past week we have had an opportunity of seeing un test a set of reduction gears for a steam turbine, just completed for Messrs. Workman, Clark & Co., of Belfast, by the Power Plant Company of West Drayton, Middlesex. This firm may not unfairly claim to be the pioneers in this department of engineering, having cut the gears supplied to the *Vespasian*, which demonstrated the possibilities of the system. The gears for Messrs. Workman, Clark & Co. are designed to transmit 2,200 horse-power with a reduction ratio of 10 to 1. The pinion is of 7.66-inch pitch diameter, and has twenty-three double helical teeth, and runs at 3,000 revolutions per minute. The wheel is of 77.33-inch pitch diameter, with 229 teeth, the useful width of the face being 3 feet, and the width over all 4 feet 6 inches. From the foregoing, it appears that the pitch-line speed is 100 feet per second, and the pressure 336 pounds per lineal inch of effective width of wheel. The test was made running light, the pinion shaft being driven by a 30-horsepower motor. The Power Plant Company have, we should add, supplied the plant complete, inclusive of its housing, oil pump, oil filter and oil cooler. The total floor space occupied measures 9 feet 2 inches by 9 feet 10 inches, and the center line of the shafting is 2 feet 9½ inches above ground level. The gear ran under test with very little noise. The lubrication is forced throughout, the oil being supplied under pressure by a small centrifugal pump arranged at one end of the pinion shaft. All the bearings are accessible without removing the main cover. The brasses are lined with white metal. The supply of oil to the pinion enters the center of a long sleeve extending from end to end of the pinion, and cut away on one side so as to clear the teeth of the driven wheel. At each end this sleeve is bored to fit the external diameter of the pinion for a length (axially) of about 1 inch; but the intervening space is larger than the pinion diameter, and this annular space is kept filled with oil by the pump. A uniform distribution of lubricant over the whole length of the teeth is thus secured. The pinion is of nickel-steel specially heat treated. The teeth of the driven wheel are cut in forged-steel rings shrunk on to cast-iron center-pieces, machined all over and accurately balanced. The above gear, large as it is, by no means represents the full capacity of the works. A set recently sent out weighed 25 tons complete. The teeth of these gears are hobbed on special tools designed and constructed at the works. The largest of these machines is capable of cutting wheels up to 13 feet 6 inches in diameter by 80 inches width of face. It is, however, probable that the Power Plant Company may ultimately abandon the hobbing process for these large gears in favor of a method of "generating" the teeth. Small double-helical wheels are already being made with generated teeth, and

with such excellent results that a large machine operating in the same way is now under construction. The new machine is a very ingenious one and has the advantage that the dividing gear has to take very much smaller pressures than when the teeth are formed by means of a hob.—"Engineering," October 1, 1915.

THE SURFACE CONDENSER.*

By C. F. Braun.

The primary functions of a surface condenser are to reduce back pressure on the exhaust side of a steam prime mover; to conserve and return to the steam generator, in the water of condensation, as many units of heat as possible, and to remove air from the feed water, thus reducing pitting of boilers. In accomplishing these results the condenser must handle four separate fluids, viz: steam, air (including other non-condensable vapors), water of condensation and cooling or circulating water. The desirable state of these fluids is not the same, the conditions to be approached being:

(*a*) *Steam* should enter the condenser, be conducted freely to all parts with the least possible resistance, reduced to the lowest practical temperature (and consequently pressure), and converted into water.

(*b*) *Air*, a non-conductor, should be rapidly cleared from the heat-transmitting surfaces, collected at suitable places, practically freed from entrained water and water vapor, and cooled to a low temperature for removal at minimum volume, with consequent least expenditure of mechanical energy.

(*c*) *Condensate* should also be rapidly cleared from the heat-transmitting surfaces, freed from air, collected at suitable points for removal, and returned to the steam generator at the maximum practical temperature.

(*d*) *Circulating water* should pass through the condenser with least friction, deposit a minimum amount of precipitated chemicals or debris, and absorb a maximum amount of heat.

The main principle of the design of the condenser is the transference of heat from the steam through the dividing surface to the cooling water, and the transfer per unit of area or of size is a measure of the efficiency of the apparatus, this being directly proportional to the temperature difference or head. For obtaining mean values of temperature differences the following formula, developed mathematically by Grashof, has been proved accurate:

$$M = \frac{D_1 - D_2}{\log_e \frac{D_1}{D_2}}, \quad \ldots \ldots \ldots \quad (1)$$

where

M = mean temperature difference,
D_1 = temperature difference between fluids at beginning,
$= TS_1 - TW_1$,
D_2 = temperature difference between fluids at end,
$= TS_2 - TW_2$,
TS_1 = initial temperature of steam,
TS_2 = final temperature of steam,
TW_1 = initial temperature of circulating water,
TW_2 = final temperature of circulating water.

It is commonly assumed that TS is constant throughout the condenser, by which this formula reduces to

$$M = \frac{TW_2 - TW_1}{\log_e \frac{TS - TW_1}{TS - TW_2}}, \quad \ldots \ldots \ldots \quad (2)$$

* Abstract of paper read before the Spring Meeting of the American Society of Mechanical Engineers, June, 1915.

but since the frictional drop through the steam space of a condenser is usually 0.5 inch or more, representing with high vacuums a temperature difference of, say, 10 degrees F., the use of formula (2) for applying to large condensers data obtained on smaller ones, or for analyzing the performance of a condenser or various sections of a condenser, will lead to serious errors. With any given set of temperature values the mean temperature difference can only be varied by arrangement of heating surfaces. These must be such as to produce counter-current flow, the circulating water entering where the steam is coolest and leaving where it is hottest.

Transfer of heat through a unit of condenser-tube area per unit of mean temperature difference was early recognized as varying greatly under different conditions, the most apparent variation being an increase with increase of water velocity. With regard to resistance the transfer of heat produced by the temperature head is opposed by a total resistance R, which for analysis divides conveniently into the resistance R_v on the vapor or steam side of the surface, the resistance R_m of the metal walls of the surface, and the resistance R_w on the cooling-water side of the surface. A simple equation expressing heat transfer in useful terms may be written as follows:

$$R = \frac{M}{H},$$

in which

H = number of heat units transferred per unit time,
M = mean temperature difference,
R = total resistance = $R_v + R_m + R_w$.

Even with high steam pressures and with superheat, the total B.t.u. to be extracted by the condenser may safely be assumed as 1,000, and it is convenient to adopt an arbitrary resistance unit such that

$$H = \frac{1{,}000 \times M}{R} \text{ or } W = \frac{M}{R}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which

H = B.t.u. per square foot per hour,
M = mean temperature difference in degrees F.,
R = resistance per square foot of surface,
W = pounds steam condensed per square foot per hour.

The symbol U will be used when M is unity, so that U = B.t.u. per square foot per hour per degree mean temperature difference. This resistance R may also be expressed in terms of equivalent conductivity by the equation:

$$R = \frac{1{,}000 \times L}{C \times 4{,}290},$$

in which

L = thickness of substance in inches,
C = conductivity in c.g.s. units

The results of the tests by Orrok, as shown in Fig. 1, curve A, provide the most reliable data available on heat transfer through condenser tubes; and in Fig. 2, curve A, are plotted total resistances R obtained by applying the values from Orrok's curve in Fig. 1 to equation (3) in which M is taken as unity. These resistances are plotted against reciprocal velocity instead of against velocity. A reasonable value for $R_v + R_m$ for Orrok's test is 0.4, and on this assumption curve B, Fig. 2, is plotted showing the relation R_w to the reciprocal velocity.

A curve, Fig. 1, curve B, plotted from the values on curve B, Fig. 2 represents the relation of heat transfer from the surface of a condenser tube to velocity of the water in contact with that surface. From this curve U_w varies directly with V according to the equation:

$$U_w = 245 + 141\ V.$$

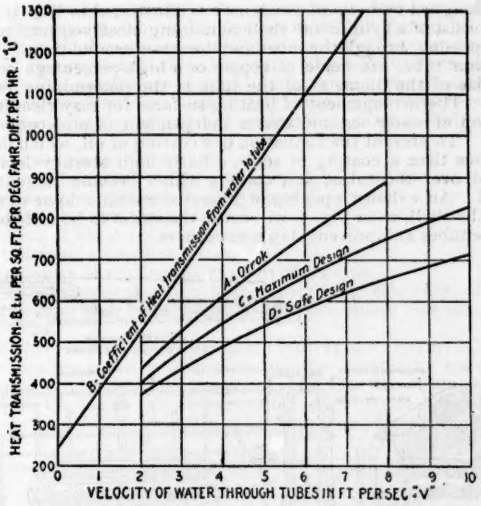

HEAT TRANSMISSION-B.t.u. PER SQ.FT. PER DEG. FAHR. DIFF. PER HR. = "U"
VELOCITY OF WATER THROUGH TUBES IN FT PER SEC. = "V"
B = Coefficient of heat transmission from water to tube
A = Orrok
C = Maximum Design
D = Safe Design

FIG. 1.—HEAT TRANSMISSION—VELOCITY CURVES.

This variation of resistance, inversely with velocity, is due to the fact that the particles of water in contact with the surface at any instant form a non-conductor which prevents the flow of heat from particles in the body of the water to the surface of the tube. The transfer of heat is really by convection, and the more rapid the removal of the heated particles, and their replacement by cooler ones, the greater the heat transfer. With the same velocity this transfer of particles is much more rapid in a small tube than a large one, where, so to speak, a cold core of water exists. The desirability of small tubes is thus indicated, and experience shows that a ½-inch to ¾-inch diameter should be a maximum.

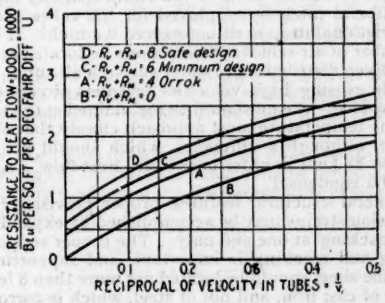

RESISTANCE TO HEAT FLOW = 1000 / B.t.u. PER SQ.FT. PER DEG. FAHR. DIFF. = U
RECIPROCAL OF VELOCITY IN TUBES = 1/V
D - Rv = Rw = 8 Safe design
C - Rv = Rw = 6 Minimum design
A - Rv = Rw = 4 Orrok
B - Rv = Rw = 0

FIG. 2.—RESISTANCE—RECIPROCAL VELOCITY CURVES.

A well-designed cylindrical condenser is illustrated in Fig. 3. Most condensers consist of a cylindrical shell containing closely-spaced round tubes, the water passing through the tubes and the steam around them. Practically all condenser tubes are made of copper or a high-percentage copper alloy, and the size of the diameter of the tube is the determining factor in its thickness. The arrangement of heating surfaces for easy cleaning, and the construction of water-channel covers independent of pipe connections, are important. To prevent the formation of a coating of oil, which has an effect more serious than a coating of scale, a fairly high steam velocity must be maintained over the tubes, and corners which become stagnant must be eliminated. An exhaust opening of liberal size with a dome extending the length of the shell, as in Fig. 3, will cause the steam to be distributed to the ends of the tubes and prevent stagnant corners.

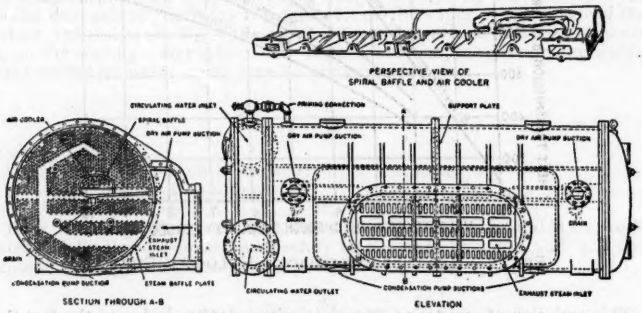


FIG. 3.—CYLINDRICAL CONDENSER OF GOOD DESIGN.

Baffle plates for directing the steam to remote parts of the condenser introduce resistance to steam flow and should be avoided, except in the case of the small plate directly in front of the exhaust inlet which protects the tubes from entrained water in the exhaust. The steam passing over the tubes condenses and diminishes in volume as it progresses, but the steam-flow velocity may be maintained by constructing a gradually reducing steamway, or by gradually reducing the pitch of the tubes, or by making lanes or passages to various parts of the steam space by omitting tubes (see Fig. 3). If a liberal pitch be employed for the tubes and ample lanes be provided, the frictional drop need not exceed 0.4 inch. It is important that a sufficient number of air-removal connections be located at points where air accumulates. Even distribution of water through all tubes is necessary, and narrow channels causing high velocities or inlets directing water on to the tubes must be avoided. Steam should be maintained at the lowest practicable pressure, and the temperature must approach closely that of the circulating-water discharge, although a difference, which should not be more than 10 degrees F., must be kept in order to produce heat flow. An air cooler is an essential part of a condenser.

Regarding general structural features, proper provision for the accommodation of expansion strains may be accomplished by expanding the tube into one head and packing at one end only. The proper supporting of tubes to prevent sagging and cracking is important, and supporting plates drilled true with the tube sheet should be located not more than 8 feet apart. Shells should be made of cast iron, and not of steel, which is corroded by the gases contained in the steam, and water channels and covers should be separate castings, so that the tube ends may be readily exposed for cleaning without

breaking the pipe connections. The connections between, and relative location of, condensers and auxiliaries are important factors in condenser efficiency, but in most cases are beyond the control of the manufacturer and consequently are neglected, thus becoming a constant source of trouble. The exhaust pipe, condensate piping, and air-pump piping should all be amply large, although the last is ordinarily much larger than necessary.

A comprehensive rating of the surface condenser must include the following:

(*a*) Quantity of steam condensed.
(*b*) Vacuum obtainable (corrected to 30-inch barometer).
(*c*) Temperature of available cooling water.
(*d*) Cooling-water exit temperature.
(*e*) Condition of air at point of removal.
(*f*) Friction head on cooling water.
(*g*) Temperature of condensate at point of removal.

The first four items express the heat-transmitting efficiency, and can, for purposes of comparison, be reduced to B.t.u. per square foot per degree difference per hour.

Since a higher vacuum at the air pump than at the exhaust inlet is of no value, the mean temperature difference used for comparing results on condensers should be computed on the assumption that the steam temperature throughout the condenser is that due to the vacuum at the exhaust inlet, and equation (2), the values of which can be determined from Fig. 4, should be used.

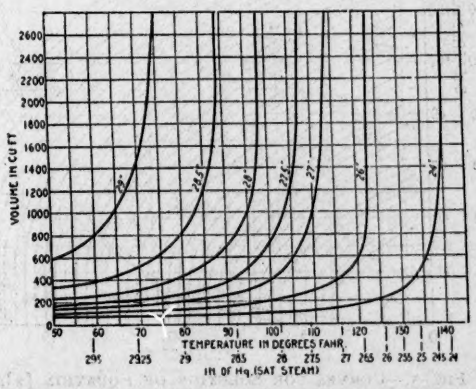


FIG. 4.—CURVES SHOWING THE VOLUME PER POUND OF AIR MIXED WITH SATURATED WATER VAPOR, FOR VARIOUS TEMPERATURES AND PRESSURES OF THE MIXTURE.

A diagram for the rapid determination of mean temperature difference values is given in Fig. 5. Probably the most reliable formula for determining these is that of Grashof, which, when the temperature of one of the mediums is constant, reduces to

$$D = \frac{T_2 - T_1}{\log_e \frac{T_s - T_1}{T_s - T_2}},$$

where D equals the mean temperature difference, T_1 the lowest temperature of the fluid, T_2 the highest temperature of the fluid, and T_s the temperature of

the gas. The result will not be changed if any constant be deducted from all the T's in the equation. By making this constant T the equation is simplified to one of three variables, which are plotted on the diagram, Fig. 5. To increase the accuracy or range of the diagram two scales are given: for

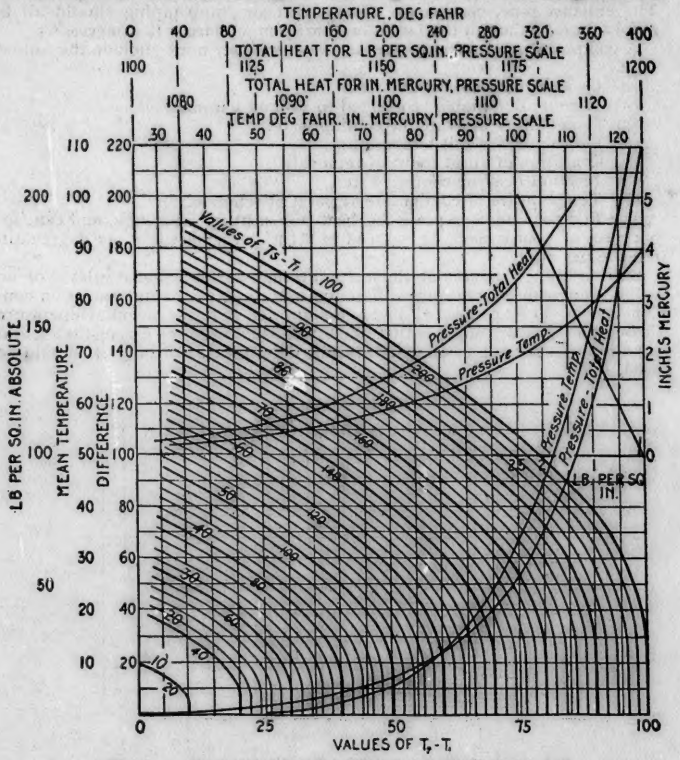

TEMPERATURE, DEG FAHR
TOTAL HEAT FOR LB PER SQ IN. PRESSURE SCALE
TOTAL HEAT FOR IN. MERCURY. PRESSURE SCALE
TEMP DEG FAHR. IN. MERCURY. PRESSURE SCALE
LB PER SQ. IN. ABSOLUTE
MEAN TEMPERATURE DIFFERENCE
INCHES MERCURY
LB. PER SQ IN.
Values of Ts - T1
Pressure-Total Heat
Pressure Temp.
VALUES OF T2 - T1

FIG. 5.—CURVES FOR SOLUTION OF EQUATION [2].

temperatures between 200 degrees and 1,000 degrees use the 100 scale. Five curves showing pressure-heat temperature relations of steam are produced on the same set for convenient reference.

The complete equation for the condenser is

$$S = \frac{W \times Q}{M \times U} \text{ or } U = \frac{W \times Q}{M \times S},$$

in which

U = B.t.u per square foot per degree F. difference per hour,
= coefficient of heat transmission,
M = Mean temperature difference in degrees F.,
W = Pounds of steam condensed per hour,
S = Square feet of cooling surface,
Q = Total heat removed by circulating water per pound steam condensed (usually taken as 1,000 in all cases for simplicity).

TABLE I.

	A	B
Square feet surface	2,000	5,500
Pounds steam condensed per hour	18,500	57,200
Vacuum, inches	28½ (1½ abs.)	27
Temperature entering cooling water, degrees	65	60
exit cooling water, degrees	80	100
Mean temperature difference per equation (1), degs.	18.3	30.9
B.t.u. per square foot per degree difference per hour	505	337

Thus, as an example, if it be assumed that the results in Table I are obtained from condensers A and B, A will be 50 per cent. more efficient than B. Items 3 to 6 in Table I determine the mechanical energy required by air condensate and circulating pumps, and item 7 indicates the heat efficiency, being a measure of the heat returned to the system in the condensate.

—"The Shipbuilder."

SALVAGE OF THE *F-4*.

NAVAL CONSTRUCTOR J. A. FURER.

Up to the time the *F-4* was lost, the United States Navy had been entirely free from serious submarine accidents. In fact, our Navy was the only one operating any considerable number of under-water craft which had not lost one or more such vessels.

The four submarines of the "F" class had been operating in Hawaiian waters since August, 1914. On the morning of March 25th, 1915, at about 9:15, the *F-4*, preceded by the *F-1* and *F-2*, stood out of Honolulu Harbor for a submerged run. On the way out, the *F-4*, running submerged, but with periscopes showing, passed the *F-1*, standing in on the surface. The commanding officer of the *F-1*, noticed that the periscopes of the *F-4* were trained on him and waved his cap in acknowledgment. That was the last signal exchanged by the doomed vessel with the living world.

The other two submarines returned to the harbor shortly, but when the *F-4* had not returned by 10:30 A. M., the officer of the deck of the tender *Alert* became alarmed and reported the fact to the commanding officer. A fast motor boat was promptly dispatched to search the harbor fairway for any signs of the vessel. The other three submarines and all the power boats of the *Alert* presently joined in the search. In the meantime, also, men had been stationed to listen in on the submarine signal sets of the *Alert* and of the other vessels. This watch was kept for days, but not the slightest sign of life was ever received from the lost submarine.

At about 12 o'clock one of the vessels came across quantities of air bubbles and an oil slick on a range approximately between Diamond Head and Barber's Point. The chart shows 1,200 feet of water in this locality. Had this spot been the resting place of the *F-4*, there would have been no use in attempting to bring her to the surface. However, the oil seemed to come from inshore, which left some hope that the vessel might not lie beyond the depth of human assistance.

The ocean floor rises very steeply to form the Island of Oahu, as it does around all of the islands in the Pacific Ocean which are of volcanic origin. Only a few hundred yards inshore from the locality covered by oil, soundings gave a depth of about 300 feet. This was encouraging and left a chance that the vessel might not be in the very deep water indicated by the first bearings.

Two divers from the submarine flotilla made a descent in this vicinity

to a depth of 190 feet without reaching bottom and without seeing anything of the *F-4*. These men went down under helmets only—this is a matter of choice, because of the somewhat greater facility in coming to the surface in case of an accident to the air supply. This depth was later exceeded by one of the divers who descended to 215 feet under a helmet alone, but again without reaching the bottom or seeing the vessel. A descent to this depth, with air supplied by a pump, is probably a record. The return of this diver without suffering from acute caisson disease or other physical impairment is almost miraculous.

The only possible chance of saving any lives—if the men survived the accident at all—was to drag the craft into shallower water. This was known to be feasible only in case the living and machinery spaces of the submarine had not been filled with water. With the vessel completely waterlogged, her weight would have been more than 260 tons. This was no doubt actually the condition of the submarine within a few hours after the accident. There was, of course, not the slightest possibility of dragging any such weight in shore and up hill. To have made no attempt at dragging would, however, have meant giving up all hope of saving the lives of the crew at once, because no lifting gear could be improvised and made fast at a depth of 300 feet, within the time available for rescuing the men.

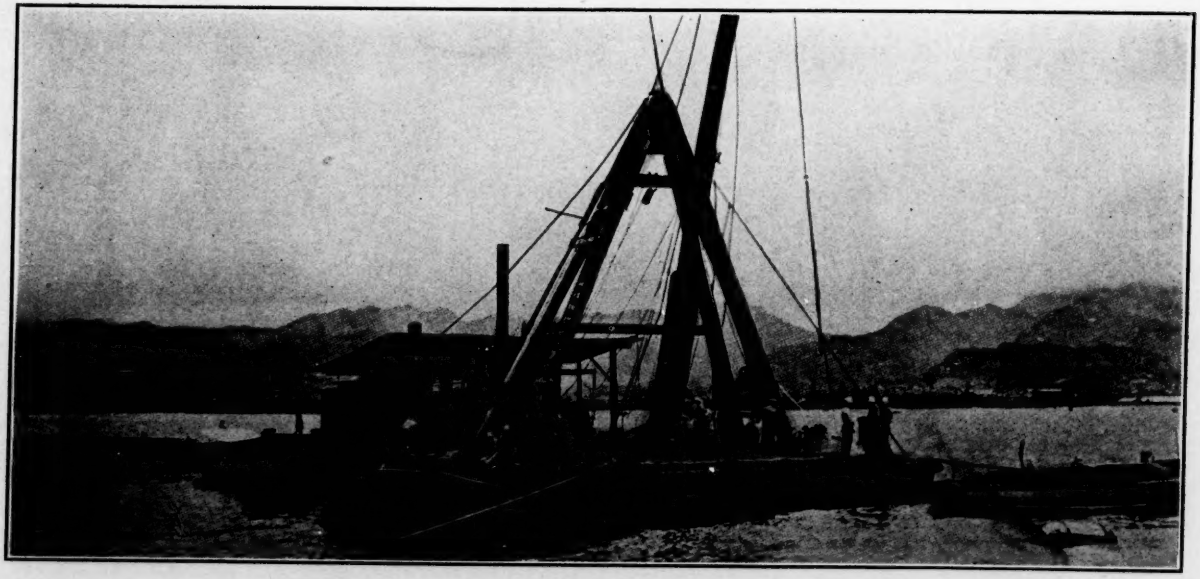
Tugs were brought to the scene of the accident to drag the bottom. For this purpose, a sweep about 2,000 feet long was made of chain and wire cables. Two tugs towed at each end, parallel to each other, and about 400 feet apart. On account of the great depth of the water, a very long sweep had to be used to insure its dragging the bottom. It was difficult to determine, even approximately, the location of the submarine, because the air bubbles showed only intermittently and the oil slick soon spread over a very large area.

During the night a microphone was constructed on the *Alert* for sounding the bottom and detecting the vessel in case of contact. This apparatus consisted of two sounding leads strapped together. Into the bottom of each lead a nail was driven. Electric conductors were connected to the nails and fitted to a telephone receiver. A number of dry cells were wired in to complete the outfit. On bridging the two nails with a metallic object, the circuit through the telephone receiver was completed—recording a click.

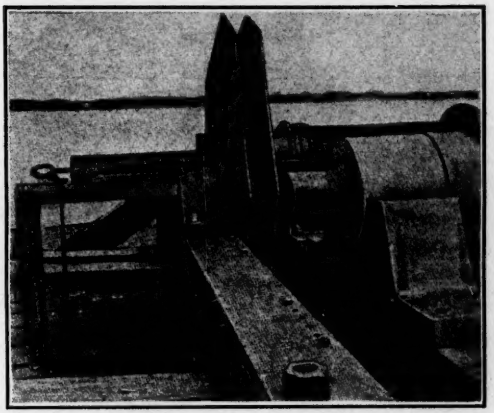
After dragging the sweep back and forth over the bottom all night, it caught firmly under the vessel about noon the next day. A good many strikes had been made previously, some of which may have been around the top of the submarine, but slipped off. The only evidence that the vessel had been caught was that air bubbles again appeared in the locality where they had been previously noted. Also, by sounding with the microphone over this area, a large metallic object was located which was correctly assumed to be the submarine. The depth of water around this spot varied from about 295 feet to 315 feet. The position was plotted and the vessel found to be lying about a mile and a half from the harbor.

All efforts to drag the vessel in shore were, however, fruitless. Even after applying an upward pull of about 100 tons, by means of a dredge, and towing in shore with all the power available, the submarine could not be moved. This was the final confirmation of the belief that the vessel must be completely filled with water. While these attempts at rescuing the crew were being made, steps were being taken to salvage the vessel by lifting. No ready-at-hand salvage gear of any description was to be had in Honolulu. This was not, however, a particular handicap, because none of its usual salvage methods could have been applied in any case, on account of the great depth of water in which the vessel lay.

For the main units of the salvage plant, two substantially-built mud scows, 104 feet long by 36 feet beam, were selected. The use of the



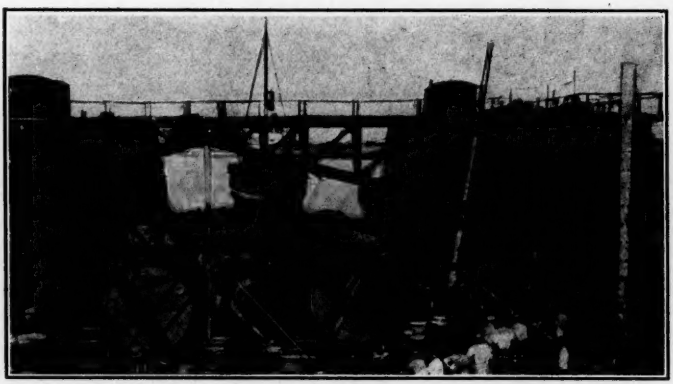
Pontoon Alongside the Wrecking Scow Ready to be Sunk.



WINDING SPOOLS OF WINDLASS—LOCKING PIN AT LEFT.

F–4 COMING UP, BOW PONTOONS ON SURFACE, MIDDLE PONTOONS JUST EMERGING.

TOWING THE F–4 SUSPENDED FROM THE PONTOONS.



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THE F–4 AND PONTOONS IN DRY DOCK.

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THE LARGE HOLE TORN BY THE SLINGS THROUGH THE SHELL PLATING.

scows strictly as pontoons—that is, the employment of their buoyancy alone for lifting—was ruled out, because there is practically no tide off Honolulu.

Windlasses were improvised instead, for lifting. Five 30-inch I-beams were first of all laid across the mud pockets so as to transfer the large concentrated loads to the fore-and-aft framing of the scows. Fortunately, heavy shafts could be procured in Honolulu, as they are used in the sugar mills, and are consequently carried in stock by the machine shops. Shafts 13 feet long by 14⅝ inches in diameter were selected for the windlasses—the lifting slings being wound up directly on the shafts. These shafts were supported by three cast-iron pillow blocks secured to the I-beams. Two windlasses were fitted side by side on each scow so that two slings might be used under each end of the vessel. It would have been quite impossible to obtain a single wire hawser of sufficient strength to carry one end of the submarine under the live-load conditions to which the slings were subjected.

A hoisting engine was installed on each scow for winding up the slings. These engines were, naturally, not powerful enough to do the work without a large purchase. In the case of one scow, the power had to be multiplied 18 to 1, and, in the case of the other—which had a smaller engine—36 to 1. This was accomplished for each windlass by reeving up a triple-sheave tackle with ¾-inch wire rope leading to one of the engine drums. To the traveling block of the tackle the winding spool rope (as shown in one of the photographs) was shackled. This rope was wound up by hand on a spool which was driven on to the square end of the windless shaft. By unwinding this rope by power, the requisite turning effort was applied to the windlass shaft. As soon as the two blocks of the triple purchase came together, the spool rope was unshackled from the traveling block—rewound on the spool by hand, the triple purchase overhauled, and again shackled to the rope, ready for another lift. During this fleeting process the windlass shaft had to be kept from turning backward. This was accomplished by slipping a nickel-steel locking pin through a hole in the spool—as shown in one of the photographs—and backing down until the ends of the pin rested on pillow blocks provided for the purpose. Four locking-pin holes were bored in each spool.

The construction of the winding spools was one of the most bothersome features of the design, because of lack of manufacturing facilities. A steel-casting plant was not available. Cast iron could not be trusted, as the stresses to which the spools were subjected in service were very great. They were therefore built up solid of steel plates. The core was made of ten ½-inch plates in the absence of any heavier available material, the side plates being of ¾-inch material. All were riveted together and the core lined with Douglas fir. As the diameter of the spools was 50 inches and the diameter of the windlass barrel 15 inches, a multiplication of power of better than 3 to 1 was obtained.

The ends of the hoisting cables were secured by means of hook bolts passing through holes in the windlass shafts. These bolts were 2½ inches in diameter. By setting up on the nut at one end of the bolt, the hook was drawn down over the cable so hard that friction prevented the end from slipping through. The end was also backed up by cable clamps and three dead turns of the lifting hawser allowed before taking up the load. The length of the scows limited the travel of the running block of the triple purchase to about fifty feet. This consequently limited to three turns the number of revolutions of the windlass shafts obtainable without rewinding the spool rope. The windlasses then had to be locked, the blocks overhauled and the rope rewound. This was the condition on the scow having the more powerful engine. On the other scow, on account of the introduction of a luff to increase the purchase, only one-half turn of the shaft could be obtained between fleets. While this method of

lifting promised to be very slow when the design was evolved, it developed to be an insignificant item in the total time consumed in the salvage operation. On one occasion the submarine was lifted 58 feet in less than two hours after all the preliminary work of preparing to lift had been completed. This preliminary work took 21 days.

In designing the gear, shafting and supports, the live-load conditions due to the almost continual swell had to be reckoned with. If the assumption could have been made than 260 tons of dead load would be distributed equally among four lifting slings, the design could have been much simplified. Instead of that, the assumption had to be made that half the weight of the vessel would come as a live load on a single sling at times. That this assumption was not extravagant was fully demonstrated during the salvage work.

Two steam-driven pumps were installed on each scow, so that in case it should become necessary to use more power for starting the submarine off the bottom than could be delivered by the hoisting engines the buoyancy of the scows could be called on. The buoyancy of the scows had to be called on only once, however.

A dredge, from which the boom and bucket gear had been removed, was used as the central unit of the salvage plant. This dredge was selected because it was equipped with a 10 x 14-inch engine geared to six drums, any of which could be operated independently of the others. This dredge was anchored near the submarine. After sweeping the lifting hawsers under the submarine, the ends were taken to the dredge and hove in by means of the drum wires—thus mooring the dredge over the vessel. The gearing between the engine and the drums was such that a very considerable pull could be applied to the hawsers, as was frequently necessary in order to make the lines render under the hull of the submarine to the desired position.

Search was made, while the scows were being rigged, for hoisting cables of requisite strength to carry the loads. Cables of these dimensions are not carried in stock by manufacturers, and, as was to be expected, had to be picked up here and there. Only one cable could be located which was entirely satisfactory. This was a plough-steel wire 2⅝ inches in diameter. For the other three slings galvanized steel cables 2½ inches in diameter, and plough-steel 2 inches in diameter had to be used. One of the chief difficulties in handling hawsers of such size is their weight and stiffness. The 2⅝-inch plough-steel cable weighed over seven tons and was not much more flexible than a bar of iron.

Within a few days after the accident the Navy Department started a party of divers from New York for Honolulu to assist in the salvage operations. The Bureau of Construction and Repair had been conducting experiments for some time on deep-sea diving. Even though the divers could do very little work at a depth of 300 feet, their services were needed to report on the position of the lifting hawsers so that they could be shifted intelligently to the desired locations.

Contrary to the general belief, a special suit is not worn for deep-sea work. The diver is subjected to the full pressure corresponding to the depth in which he is working, as is the case in ordinary shallow-water diving. The difference between the deep-sea procedure and the shallow-water procedure lies merely in the method of supplying air, and in the scientific restoration of the diver's body to a normal state of equilibrium while returning to the surface after having been subjected to abnormal pressures. Air for deep-sea work is supplied from storage tanks—the pressure being regulated from above, according to the depth at which the diver is working. In this instance a number of torpedo flasks were charged to a pressure of 2,250 pounds. The air is supplied to the diver's helmet through the usual hose line after being passed through a reducing valve. The diver also has control valves so that he can regulate the air

supply. By means of the helmet valves he can make himself lighter or heavier at will. When the submarine was in 300 feet of water the pressure to which the diver was subjected when on the bottom was about 135 pounds to the square inch.

The rate at which the diver is brought to the surface is fixed by the length of time he was exposed to any given pressure. For example, the first diver to go down on this job made the descent in five minutes and was on the bottom at a depth of 306 feet for twelve minutes. He was then brought back to the surface in one hour and forty-five minutes to insure gradual decompression.

The diver is allowed to come up fairly rapidly until a depth of 100 feet is reached. A Jacob's ladder of this length, with rungs spaced 10 feet apart, is sent down to him so that when he reaches this depth in his ascent he can rest on the lowermost rung. The diver is required to stop on each rung for the length of time called for by certain tables which have been worked out from experimental data. Decompression is in this manner accomplished slowly enough so that he will arrive at atmospheric pressure restored to normal condition.

Although the diver does not become conscious of fatigue while working under great pressure much sooner than if he were working near the surface, the exertion is, nevertheless, much more exhausting. This was demonstrated in the case of a diver who remained on the bottom 30 minutes, trying to pass a small reeving line—the only actual manual work at this depth which was attempted. According to the diver's statement, he was not conscious of becoming fatigued while on the bottom, but after coming to the surface he collapsed from exhaustion and did not regain his strength for three or four days, notwithstanding the fact that the man was a perfect physical specimen.

After innumerable difficulties the four wire hawsers were placed under the submarine in the desired locations and the ends made fast to the windlass shafts. As was feared, however, the strength of the cables became impaired rapidly, owing to chafing, where they passed around the bilges of the vessel. After lifting about 25 feet and towing inshore to a new landing one of the smaller hawsers parted. While it was being replaced bad weather set in, and presently all of the other hawsers parted likewise. Just one month from the date of the accident not a line was around the vessel and the only measurable progress was a reduction in the depth of water from 300 to 275 feet.

However, the adequacy of the lifting gear had been demonstrated as well as the feasibility of sweeping lines under the vessel. It was now obvious that wire hawsers could not be counted on to resist the chafing action caused by the more or less continuous swell. Four slings were made up instead, of 90 feet of 2⅝-inch chain into the ends of which the wire hawsers were spliced. By having only chain in contact with the hull of the vessel the stranding of the wire hawsers was avoided. Chain causes more damage to the plating of a vessel when so used than wire. For this reason that type of sling was ruled out originally. Great difficulty was experienced in getting the new slings in place, as the former method of sweeping by means of two tugs was impracticable, but they were finally worked to the desired locations.

The vessel was lifted 225 feet in a few days, once the slings were in place, and towed inshore to a spot just off the channel entrance where the depth of water is only 50 feet. Preparations were being made for a final lift which would have brought the submarine high enough to permit entering the channel and landing the vessel in a floating dock in the harbor. Only a few hours were needed for this work, but in that short space of time a heavy ground swell set in and wrecked the operations. High surf waves built up in an incredibly short time which caused the scows to charge back and forth so violently that the shell plating of the

submarine gave way under the forward slings. All the lines had to be let go in order to save the vessel from becoming a complete wreck and the scows from being washed up on the reef. The ground swell was followed shortly by heavy southerly weather. When, several days later, the sea had moderated sufficiently, divers were sent down to examine the condition of the vessel. A large rent was found in the hull forward where the chains from the leading scow had ripped through the shell plating. Lifting by the windlass method was now unsafe, as it was likely that the forward end of the submarine would break off under the strain.

A new plan was then adopted consisting of the use of six specially constructed pontoon cylinders. All of the cylinders were 32 feet long—four being 11 feet in diameter and two 12 feet 6 inches in diameter. The combined lifting capacity of the six pontoons was 420 tons. A margin of 160 tons was allowed for breaking the vessel away from the bottom, should there be considerable sand suction, and for the failure to realize all of the buoyancy available which would be occasioned by some of the cylinders becoming cockbilled to such a degree as to make it impossible to blow all of the water out of the high ends.

Each cylinder was divided at midlength by a transverse, water-tight bulkhead so as to facilitate control in sinking. A 4-inch flood and discharge valve was installed in each end bulkhead close to the bottom. Air valves were fitted on top of the cylinders for blowing out the water and venting each compartment. Eight feet from the ends, 12-inch hawse pipes were fitted through the pontoons.

The plan called for lifting the vessel by means of six chains—the ends of the chains being brought up through the hawse pipes of the pontoons, three of which were to be placed on each side of the vessel. The chains were first of all worked under the submarine in predetermined positions—two forward, two amidships and two aft. The two chains of a pair were spaced 16 feet apart so as to correspond to the distance between the hawse pipes. The pontoons were planted by means of a wrecking scow which was moored accurately over the submarine. The two cylinders of a pair were towed to the scene of operations and placed one on each side of the scow, directly over a pair of chains lying on the bottom.

The ends of the chains were now fished up through the hawse pipes and the cylinders submerged by opening the flood valves. They were kept under control while sinking by means of 5-inch manila lines made fast to the ends and taken to the hoisting engines on the scow. The process therefore consisted of threading the pontoons on to the chains. After they had landed on the bottom the chains were adjusted so as to leave the necessary amount of slack to permit the pontoons to rise just clear of the vessel on becoming buoyant. A clamp was now fitted to each chain by the divers, just above the hawse pipe. The clamps consisted of two steel castings, molded to the shape of the chain links and of such length that they spanned the hawse pipes. By means of four heavy bolts the two halves of the clamp were drawn together. These bolts kept the clamps from spreading and the chain from slipping through when subjected to the lifting strain.

After all six cylinders had been landed on the bottom alongside of the vessel and the clamps had been secured by the divers, the unwatering process was started. For this purpose torpedo air flasks, charged to a pressure of 2,150 pounds, were placed on a coal barge. The flasks were piped to an expansion chamber which in turn was connected to a manifold. Twelve ¾-inch air-hose lines were led from the manifold valves to the blow-out valves on top of the pontoons. The coal barge was moored about 50 feet to one side of the location of the submarine so as to be clear when the latter rose to the surface. A pressure of 35 pounds was used at the manifold for blowing out the water while the cylinders

were resting on the bottom. The unwatering operation had to be watched very carefully so as to avoid the danger of blowing out the heads of the cylinders on emerging.

The method proved successful in every detail. The work of placing the chains under the submarine was started on August 21st. All of the chains were in position on August 25th. It then took one day to place each pair of cylinders. This could be done only during daylight, as, naturally, the divers could not work in the dark. On August 29th everything was ready for blowing the water out of the pontoons. This operation took about two hours from the time the air was turned on. One end of the submarine came up slightly ahead of the other, as was to be expected. The vessel was towed into the harbor suspended from the six pontoons, and was docked the following day.

The pontoon method which was finally adopted could not have been used originally for bringing the vessel to the surface because of the great depth of water in which the submarine lay at first. The divers had to work under water from five to nine hours a day from August 21st to August 29th while placing the chains under the vessel, securing the clamps after the pontoons had been lowered, closing flood and vent valves, and, later on, connecting the hose leads for blowing out. This work could not have been done at a depth of 300 feet, as most of it was extremely arduous and fatiguing.—"Scientific American."

GALVANIC CORROSION DAMAGES HULL OF YACHT.

A very large new sailing yacht, the *Sea Call*, built for Alexander Smith Cochrane, of Yonkers, N. Y., at a cost said to have been in the neighborhood of half a million dollars, has been so injured by corrosion, within three months after her launching, that she has been dismantled and scrapped. The case is of especial interest to engineers, as it is one of the most remarkable instances on record of serious damage resulting from the galvanic action of dissimilar metals in a structure exposed to sea water.

The yacht was intended by its owner for ocean cruising. He desired to visit out-of-the-way corners of the world where no facilities exist for dry-docking or hauling vessels out of the water to remove marine growths from the hull. It was therefore decided to build the portion of the hull under water of monel-metal plates, using steel plates, however, above the water line. Monel metal is an alloy produced and marketed by the International Nickel Co. and named in honor of Ambrose Monell, its president. It is an alloy containing about 67 per cent. nickel and 27 per cent. copper. As might be expected from these chief constituents, the metal is highly resistant to corrosion. It has a strength about equal to that of mild steel.

According to the statement of the builders, it was the original intention of the vessel's designer, William Gardner, to use monel metal for the stem of the vessel, the rudder frame and the propeller frame, as well as for the under-water plating. Some difficulty was found in producing these parts in monel metal with the necessary accuracy and freedom from warping, and it was finally determined to use steel for these pieces instead.

The framing of the hull was of steel bulb angles. Monel-metal rivets were intended to be used in all monel-metal plates. In a very few cases, however, steel rivets were used by mistake in the monel-metal underwater plating. Attention was first attracted to the serious corrosion that was going on by the failure or total disappearance of one of these steel rivets. A stream of water entering through the open rivet hole was temporarily stopped by a pine plug and later by sending down a diver

who had put in a steel bolt as a permanent repair. In a very few weeks, however, this bolt was also eaten away. About this time it was decided to haul the yacht out of the water to remove the marine growths that had accumulated on the hull, for it was found, contrary to the expectatoin of the designers, that the salt water had so little effect on the monel-metal plates that the hull rapidly accumulated enough "grass" to check the vessel's progress. The serious corrosion which had taken place on the steel parts under water was discovered when the hull was exposed to view.

The most severe corrosion took place on the thin steel rib which forms the outer frame of the rudder (the rudder itself being covered with monel-metal plates). The reason why the corrosion was so severe here was very likely because of the rapid flow of water across the rudder edge, removing the corrosion as fast as it took place and thus presenting clean metal to the action of the electric current. A similar effect, almost as severe, took place on the steel stem of the vessel. Deep pits were eaten into the steel. There was heavy corrosion also on the steel propeller-frame casting and rudder shank.

The monel metal itself, as would be expected, showed not the slightest evidence of corrosion anywhere; and had the corrosion been confined to these outer steel pieces described, they could probably have been replaced at no serious expense by similar parts of monel metal and the galvanic action thus terminated. A more serious situation, however, was presented by the interior of the vessel. It was clearly evident, from the action on these exterior steel parts, that as the interior steel framing was connected to the monel-metal plates, corrosion would be inevitable there under the action of the bilge water. The probability that this might proceed unnoticed until the vessel's structure would be seriously weakened was doubtless the consideration that led to the decision to cut up for scrap a vessel which had been launched no longer ago than last March.

The *Sea Call* was 150 feet long on the load-water line, with a length over all of 214 feet, her bow having an enormous overhang. The beam of the vessel was 33 feet 6 inches and the draught was 18 feet. The vessel had a heavy lead keel covered with monel plates, and in addition a large centerboard. It was to be equipped with a 400-H.P. internal-combustion engine as auxiliary power, but the engine was never installed.

The yacht was dismantled at the yards of her builders, George Lawley & Sons, at Neponset, Mass. On Tuesday, August 24, the hull was hauled out of the water on the marine railway at the Lawley yards, where it was inspected by one of the editors of "Engineering News."

The possibility of galvanic corrosion occurring had evidently suggested itself to the designers of the vessel, for we are informed that experiments extending over several months were made at the Lawley yards on plates of monel metal and steel electrically connected and immersed in sea water. These experiments, we are informed, showed no appreciable corrosion of the steel and no measurable current in the wire connecting the plates, and it was therefore deemed safe to proceed with the construction of the *Sea Call* on the lines described. The present condition of the vessel, however, is indisputable evidence that the conditions in the boat itself were entirely different from those which existed in making the experiments referred to. It is doubtless true that the area of monel metal exposed to the sea water being very large in comparison with the area of steel exposed, the corrosion of the steel was concentrated.

It seems indeed strange that some such action should not have been foreseen in view of the large amount of experience on record as to the effect of galvanic action between steel and some of the copper alloys when immersed in sea water. Monel metal, we are informed, is almost identical with manganese-bronze in its electrical relation to steel when connected as a galvanic couple. In ships which use manganese-bronze propellers it

has become common practice to attach plates of zinc to the hull. As the zinc is more electro positive than the steel, the corrosion due to the galvanic action with the manganese-bronze propeller is concentrated upon the zinc and the steel is protected.

Only a few miles distant from the Neponset shipyard is the Charles River dam. On the lock gates of this dam truck axles of manganese-bronze were used. In "Engineering News" of March 18, 1909, Edward C. Sherman described the extensive tests which were carried on to ascertain the amount of galvanic action that would take place on steel and manganese-bronze joined electrically and immersed in sea water. As a result of these tests the steel portions of the dam were protected from corrosion by wrapping zinc sheets around the bronze axles.—"Engineering News."

NAVAL VESSELS.

UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1915.		Percentage hull completed Nov. 1, 1915.	
						Oct. 1	Nov. 1	Total.	On ship.
	BATTLESHIPS:								
36	*Nevada*	Fore River S. Co.	Curtis turbine	2	20.5	96.25	96.98	97.4	97.3
37	*Oklahoma*	New York S. Co.	Reciprocating	2	20.5	96.85	96.90	98.2	98.2
38	*Pennsylvania*	Newport News Co.	Cur. trb. grd. cr.	4	21	83.39	85.03	90.4	88.1
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	57.70	60.49	74.6	72.5
40	*California*	Navy Yard, N. Y.	Electric	4	21	...	...	...	...
41	*Mississippi*	Newport News Co.	Cur. trb. grd. cr.	4	21	12.14	13.31	33.1	17.4
42	*Idaho*	New York S. Co.	Pars. trb. grd. cr.	4	21	28.75	30.88	45.1	35.3
	DESTROYERS:								
57	*Tucker*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	85.82	89.14	85.7	84.7
58	*Conyngham*	Wm. Cramp & Sons	Pars. trb. grd cr.	2	29.5	87.54	91.03	88.4	87.6
59	*Porter*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	84.47	87.72	85.3	84.5
61	*Jacob Jones*	New York S. Co.	Pars. trb. grd. cr	2	29.5	93.04	94.63	92.2	92.2
62	*Wainwright*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	92.23	93.83	91.5	91.5
63	*Sampson*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	70.64	71.10	65.1	60.5
64	*Rowan*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	70.64	71.10	57.5	51.5
65	*Davis*	Bath Iron Works	Pars. trb. grd. cr.	2	30	45.92	55.68	57.3	54.0
66	*Allen*	Bath Iron Works	Pars. trb. grd. cr.	2	30	44.00	54 04	53.8	50.0
67	*Wilkes*	Wm. Cramps & Sons	Pars. trb. grd. cr.	2	29.5	48.52	51.42	34.1	28.6
68	*Shaw*	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	11.80	14.00	12.5	7.8
	FUEL SHIPS:								
14	*Maumee*	Navy Yard, Mare Isl'd	Diesel	2	14	73.82	75.31	96.8	96.5
15	*Cuyama*	Navy Yard, Mare Isl'd	Reciprocating	2	14	3.02	5.00	8.1	0.0
	SUBMARINE TENDER:								
2	*Bushnell*	Seattle Con. & D. D. Co.	Pars. trb. gearing	1	14	98.52	98.52	98.7	98 5
	DESTROYER TENDER:								
2	*Melville*	New York S. Co.	Pars. trb. gearing	1	15	98.26	99.37	99.8	99.8
	Transport	Navy Yard, Phila.	Reciprocating	2	14	12.76	23.83	36.1	33.0
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	15.88	17.74	31.1	28.5
	TUGS:								
17	*Wando*	Navy Yard, Charleston	Recip. oil fuel	1	11	...	...	...	...
18	*Pocahontas*	Navy Yard, Norfolk	Recip. oil fuel	1	11	12.00	14.00	...	...
	Ferry Launch	Navy Yard, Charleston	Reciprocating	...	...	....	....	...	...
	Freight Lighter	Navy Yard, Portsmouth	.....	...	...	90.00	90.00	...	...
	SUBMARINES:								
31	*G-3*	Navy Yard, N. Y.	Diesel-Sulzer	2	14	93.00	93.00	88.6	88.4
40	*L-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	98.9	98.9
41	*L-2*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	98.5	98.5
42	*L-3*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96.89	96.89	98.1	98.1
43	*L-4*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95.99	95.99	98.1	98.1
44	*L-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	19.90	30.70	81.0	74 5
45	*L-6*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	13.64	15.62	70.1	67.1
46	*L-7*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	13.48	14.43	67.4	63.7
47	*M-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95.43	95.56	88.7	88.4
48	*L-8*	Navy Yard, Portsmouth	Diesel-Sulzer	2	14	4.57	5.05	56.2	53.7
49	*L-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	92.53	94.22	86.4	84.7
50	*L-10*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	91.96	92.11	83.9	80.4
51	*L-11*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	88.37	90.64	80.3	75.1
52	*Schley*	Elec. Boat Co., Quincy	Diesel-New Lond	2	20	...	....	0.0	0.0
53	*N-1*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	18.75	26.19	34.3	28 8
54	*N-2*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	18.75	24.59	34.3	28.8
55	*N-3*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	18.63	23.27	34.3	28 8
56	*N-4*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	6.00	6.00	44.0	36.2
57	*N-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	6.00	6.00	41.9	35.3
58	*N-6*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	6.00	6.00	40.0	33.4
59	*N-7*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	6.00	6 00	40.4	33.7

GERMAN SUBMARINE CONSTRUCTION IN BELGIUM.

The French journal "La Nature" reproduces a statement from the Dutch newspaper "Telegraaf," according to which the keels of nine submarines have been laid down at Hoboken, near Antwerp, since March 19. The men employed in the construction of the boats number 800. The main entrance to the yard is on the northern side. Since the aerial attack carried out by the British, the Germans have taken measures against a renewed attack. Above two parts of the yard they have built a roof formed of steel plates, covered by sand bags. The yard is, further, closed in by a thick wall provided with iron doors which can be instantly closed from the inside. By these means, the boats and the men are efficiently protected against any bombardment. There is, moreover, no risk of a fire spreading to the yard, and this is important, since two very large tanks containing naphtha and engine oil are close by. The smaller of these tanks is on the northern side, the other is to the south and in the ground; it measures 8 m. (26 feet) in length, 6 m. (19 feet 8 inches) in width, and 4 m. (13 feet) in depth. Between this and the river are located two buildings, the canteen, and the ambulance. There is, further, a small bay between the "naval yard" and the Cockerill yard, which the Germans have almost entirely covered over by means of tree trunks tied together, so as to afford a passage to the Cockerill yard. They have found use for a building called "La Chapelle," in which they have mounted anti-aircraft guns. The hulls of submarines alone are built at Hoboken; the whole of the machinery and outfit is manufactured in Germany.—"Engineering."

THE JAPANESE NAVY.

Japan contemplates a new naval construction expenditure of £2,333,332 in 1915–16. For 1916–17 the estimate is £3,604,849; for 1917–18, £2,681,864; and for 1918–19, £636,047. The total expenditure proposed in four years is considerably over £9,000,000. The naval estimates for the current financial year have been approved by the Japanese Diet. The estimates will defray in part the cost of three battleships of 30,000 tons each—one, to be named the *Yamashiso*, to be built at Yokohama; another, the *Ise*, to be built at the Kawasaki yards; and a third, the *Hyuga*, built at the Mitsu Bishi yard. The estimates will further partly provide for the cost of four first-class torpedo-boat destroyers of 1,000 tons each, two of which are being built in England, as well as for the cost of four second-class torpedo-boat destroyers and two submarines of 700 tons each. The keel of the *Ise* was laid on May 5, and that of the *Huyuga* on May 11. The three battleships are to be pushed forward vigorously, and the eight destroyers and the two submarines are to be completed by the close of 1916. The two torpedo-boat destroyers which are being built in England are to be named the *Urakaze* and the *Kawakazi*.—"Engineering."

SHIPBUILDING.

Lloyd's annual register of shipbuilding for 1914–15 is a further indication, if such be needed, of the amount of wealth which the war is sending into the United States in particular, and neutral countries in general. Obviously the construction of merchant vessels in this country has fallen off very considerably owing to the demands upon our shipbuilders for vessels of various kinds for Government purposes. In some yards the construction of ships for ordinary purposes has ceased altogether, and

the trade has been diverted elsewhere. Similar conditions, of course, apply to all countries engaged in the war, hence the great prosperity in shipbuilding circles of the United States, Japan, the Scandinavian countries and Holland. In the former there is under construction for classification under Lloyd's register, at the present moment, the largest amount of tonnage on record for that country. One of the not least interesting items in the report is the information that 22 new oil vessels, having a tonnage of 120,324, have been classed at Lloyd's, and many schemes are under consideration for still further increasing the quantity of tonnage for carrying oil by converting existing vessels, and those building, into tank steamers. The lack of tank steamer accommodation had always been put forward as the reason why prices of fuel and other oils were inclined to soar. With the greater demand upon the oil resources of the world at the moment, considerable impetus has been given to increase the tonnage of this class of vessel; indeed, dire necessity has forced it. In other directions there has been a heavy demand for vessels with refrigerating apparatus, and reconstructions and modifications have been made in vessels not so provided in order to meet this demand. The total number of vessels completed and classed with Lloyd's last year amounted to 733, representing 1,715,500 tons, which is slightly in excess of the figures for the previous year; but, as already pointed out, the distribution of this among the various shipbuilding yards of the world has undergone a startling change.—"Mechanical World," Manchester, England.

AN APPRECIATION OF A WELL-KNOWN ENGINEER.

CHIEF ENGINEER BENJAMIN F. ISHERWOOD.

The August number of our JOURNAL contained a tolerably full and appreciative obituary notice of this eminent man; but there were necessarily omitted several points of great interest to marine engineers in general, and those of the United States Navy in particular, and when his life shall be adequately written, all such notes will be of value: for this reason the following additional record is submitted.

It is not generally known that Mr. Isherwood, in his comparatively early days, wrote several articles for the "Franklin Institute Journal" of Philadelphia, when that periodical was almost the sole representative in this country of scientific and technical literature, and his translations from the French ranged through a long and busy life. Shortly before our Civil War he devoted great attention to the theoretical side of steam engineering, and kept in touch with the writings of Tyndall, Joule, Clausewitz, Heimholtz, Mayer, Thomson, Pouillet and other Europeans who showed that there was more to the subject than the mere boiling of water; even in the busy times of the war he caused some of the most notable of such writings in foreign magazines to be carefully copied, solidly bound and conspicuously labelled with the then somewhat mysterious title, "Thermodynamics." Perhaps his best known contributions to marine knowledge were included, besides others, in two volumes of "Engineering Precedents," referring to various vessels, experiments with propelling instruments, comparisons of coals, the expansion of steam, description of boilers, etc., and followed by two larger volumes of "Experimental Researches," in further pursuance of the same or similar subjects. When in after years the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS was suggested, Mr. Isherwood took great interest in the project,

wrote occasional articles and translations for its pages, and, as was fitting, he was given the place of honor in the first number of the first volume.

In the trying times produced by the Civil War Mr. Isherwood was conspicuous; for while it is true that the Navy contributed its full share toward the final victory of the Union arms, it may also be claimed that the Navy owed much of its efficiency to his efforts. As each problem presented itself he helped to solve it. The first requirement was for small armed vessels for coast and harbor service, and this call was partially met by the "ninety-day gunboats;" the second step was to furnish fast vessels for action in rivers and inlets, and which, to save maneuvering in narrow waters, could proceed in either direction; this need was fulfilled to a great extent by the "double-enders;" the third demand, toward the end of the war, was for excessively speedy ships to pursue privateers; this want was sought to be redeemed by very large ships driven by the "hundred-inch engines," and at such unprecedented naval speed for those days that some foreign journals refused to believe the authenticated accounts. When certain American engineers criticised his productions (and every man who achieves distinction will be criticised), he vanquished them by allowing them similar hulls to be propelled by machinery of their own design.

Perhaps Mr. Isherwood's chief characteristic was accuracy, he spared no pains for himself or his subordinates in the pursuit of truth by comparison and experiment, and the results, however voluminous, were almost always copied for publication by infinite labor in his own neat and distinct handwriting. He appears to have had time for few amusements, but he could unbend on occasion, and thoroughly enjoyed a good opera.

He was a man of mark, and when the time shall again come of peril to this country, those who are entrusted with the design, production and working of the machinery of our Navy, can do no better than imitate the energy, industry, forethought and patriotism of Benjamin F. Isherwood.

—F. G. McK.

BOOKS RECEIVED.

SHIP FORM, RESISTANCE AND SCREW PROPULSION. A book by G. S. BAKER, for the use of Naval Architects, Engineers and Draftsmen.

The subjects are handled from a practical viewpoint and theory is only introduced when it has a direct practical bearing. In Part I are nomenclature, stream-line motion, stern-friction resistance, eddy making, waves and wave making, ship-model experiments, curves of area, etc. In Part II, entitled "The Screw Propeller," the following subjects are taken up: screw propeller, elements of propulsion, hull efficiency, wake and thrust deductions, cavitation, etc.

Diagrams illustrative of the text are included.

Published by D. VAN NOSTRAND COMPANY, New York. Price, $4.50, net.

ENGINEERING PROBLEMS, by W. M. WALLACE, contains a collection of the most important rules and data on which are based the more usual calculations in engineering work, with problems to illustrate the application of these rules.

THE TECHNICAL PUBLISHING COMPANY, LTD., 55 and 56 Chancery Lane, London, W. C., England. Price, 3 shillings, net.

HYDRAULICS, by W. M. WALLACE. A treatise for engineering students and engineers in practice.

THE TECHNICAL PUBLISHING COMPANY, LTD., 55 and 56 Chancery Lane, London, W. C., England. Price, 4 shillings, net.

ASSOCIATION NOTES.

The annual meeting of the Society was held on October 5, 1915.

The following nominations were made for officers for 1916:

For President, Captain C. W. Dyson, U. N. Navy.

For Secretary-Treasurer, Lieutenant A. T. Church, U. S. Navy.

For Members of Council:

Captain T. W. Kinkaid, U. S. N.
Engineer-in-Chief C. A. McAllister, U. S. C. G.
Naval Constructor Robert Stocker, U. S. N.
Lieut. Commander Arthur Crenshaw, U. S. N.
Lieut. Commander J. H. Tomb, U. S. N.
Lieut. Commander H. T. Winston, U. S. N.
Lieut. Commander J. O. Richardson, U. S. N.
Lieut. W. T. Conn, U. S. N.
Lieut. S. C. Hooper, U. S. N.
Lieut. W. T. Lightle, U. S. N.

The election will be held on December 30, 1915.

It was decided to submit the question of giving a banquet during the year 1916 to a vote of the members.

The following members and associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Bean, Carlos, Lieutenant, U. S. N.
Boyd, Harry L., Captain of Engineers, U. S. C. G.
Dalton, Donald McL., Ensign, U. S. N.
Downey, Thomas F., Ensign, U. S. N.
Grove, George W., Ensign, U. S. N.
Hamlet, Harry G., Captain, U. S. C. G.

Hill, George C., Ensign, U. S. N.
Kinne, Merrill T., Ensign, U. S. N.
Lafrenz, Walter F., Lieutenant, U. S. N.
Lockwood, Charles A., Jr., Lieutenant, U. S. N.
Manock, Frank D., Lieutenant, U. S. N.
Marvin, David P., 3d Lieutenant, U. S. C. G.
Meredith, John E., Lieutenant, U. S. N., Retired.
Moore, James M., Captain, U. S. C. G.
Moore, Warren L., Lieutenant, U. S. N.
Munter, William H., 1st Lieutenant, U. S. C. G.
Nichols, Newton L., Lieutenant, U. S. N.
Nimitz, Chester W., Lieutenant, U. S. N.
Parks, Charles M., Civil Engineer, U. S. N.
Picking, Sherwood, Lieutenant, U. S. N.
Riedel, Walter A., Lieutenant, U. S. N.
Satterlee, Charles, Captain, U. S. C. G.
Shoemaker, Henry E., Lieutenant, U. S. N.
Skinner, Harry G., care of Griscom-Russell Co., 90 West St., New York City.
Snowden, Thomas, Captain, U. S. N.
Sparrow, Herbert G., Lieutenant Commander, U. S. N.
Welch, Leo F., Lieutenant, U. S. N.
West, Horace B., Captain, U. S. C. G.
Wood, Horatio N., 1st Lieutenant of Engineers, U. S. C. G.

ASSOCIATES.

Ballin, A. E., care of McIntosh & Seymour Corporation, Auburn, N. Y.
Berriam, Henry C., Box 655, Newport News, Va.
Cunningham, W. P., 501 Marshal St., Hampton, Va.
Daughtrey, J. W., 5010 Huntington Avenue, Newport News, Va.
Edwards, R. H., Newport News, Va.
Everett, Harold A., M. E., Post Graduate Department, Naval Academy, Annapolis, Md.
Freeman, W. R., 2507 Parrish Avenue, Newport News, Va.
Gardner, Dan D., 91 32d St., Newport News, Va.

Gerell, John W., 216 50th St., Newport News, Va.
Gustafson, William G., care of Main Officer, Newport News Shipbuilding and Dry Dock Co., Newport News, Va.
Hope, Herbert A., 3112 West Avenue, Newport News, Va.
Lamberton, B. P., Southern Building, Washington, D. C.
Meurk, B., Newport News, Va.
Pickering, Herbert E., Navy Yard, Mare Island, Vallejo, Calif.
Powell, Charles B., Hampton, Va.
Preston, Robert, Engineer Lieutenant Commander, R. British Navy, care of Admiralty, London, England.
Rasmussen, Otto, 224 48th St., Newport News, Va.
Sugden, Robert A., Victoria Avenue and Bridge St., Hampton, Va.
Temple, J. Clarence, Newport News, Va.
Woodward, John B., care of Engine Estimating Department, Newport News Shipbuilding and Dry Dock Co., Newport News, Va.